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Note on the secondary stresses in lattice girders.

Figs. 1 to 13, pp. 278 to 283.

The mathematical investigation of secondary stresses in lattice girders has for a long time exercised the minds of many engineers in all countries. The opinion of most practical engineers is to attribute little importance to them as affecting the elastic deformation of the structures. As a matter of fact the knowledge of secondary stresses, in general, is very slight. As the calculations are laborious and tedious, few engineers take the trouble to make them and thereby clear up their ideas. The problem none the less exists, and is still receiving the attention of technical experts. In recent years, American engineers have contributed towards the solution of the question, both theoretically and by tests of bridges.

The object of the present article is to summarise the latest information on the subject available in Anglo-Saxon countries. In an article appearing in the : *Proceedings of the American Society of Civil Engineers*, September 1924, *Mr. Cecil Vivian von Abo*, under the heading *Secondary Stresses in Bridges*, deals with various methods of calculating the secondary stresses in lattice girders with rigid connections, and applies them to the particular case of a Warren girder with intermediate vertical members.

In the first place, it must be remembered that the secondary stresses are due to various causes, such as, *eccentric connections* and *rigid joints*. In any triangulated system the neutral axes of the members must intersect at the joints. If this condition is not satisfied, secondary bending moments are introduced at the ends of the members equal to the product of the principal tension multiplied by the eccentricity.

Moreover, in any triangle the primary stresses elongate or shorten the members, producing variations in the angles which are resisted by the rigidity of the joints. The fact that the angles cannot alter sets up secondary bending moments. It is mainly the determination of these moments and stresses which has exercised the ingenuity of investigators.

Mr. von Abo gives in his article a very complete bibliography. He points out that Manderla, in 1879, first shewed how the problem could be solved.

In 1877, the Polytechnic School at Munich offered a prize for a solution of the problem worded by Professor Asimont as follows : « What stresses occur in the members of a bridge truss owing to the fact that the angles of the truss triangles suffer no change. »

In 1879, H. Manderla, Assistant at the Polytechnic Schoool at Munich, submitted an excellent solution of the problem.

Since that time. Engesser, in 1879, Winkler in 1881, Landsberg in 1885, Müller-Breslau and Ritter (1890) and Mohr (1892) have published alternative solutions.

In 1902, Professor Patton of Moscow published a remarkable work on these secondary stresses. Other professors and engineers, amongst them being Isami Hiroi, of the Tokio University in 1905, and Pirlet, of Aix-la-Chapelle, have written interesting articles on this subject.

In 1913, José Páez, and in 1917, Dr. Mao, presented theses on secondary stresses, to Cornell University in the United States. These names should not let us overlook the valuable and accurate, though concise and to the point, contributions the French experts have made to the study of secondary stresses. The names of Mesnager, the inventor of the « free joint », Rabut, Dupuy and Resal are sufficient, to illustrate the part taken by French science in this investigation.

* * *

In practice, the *fundamental data on which the calculations are based* are the *angular changes* resulting from the *strains* caused by the *primary stresses.*

For any given joint the deflection angle of one of the members is selected as the unknown quantity. The deflection angles of the other members are the sum of the reference deflection angle and the changes of the angles.

These angular deflections determine the moments at the ends of the members.

The linear equations for every joint will contain the deflection angles of all the members that enter at the joint, as each joint is in equilibrium under all bending moments about the joint.

In any triangle *abc* with angles $\alpha\beta\gamma$ under unit stresses s_1, s_2, s_3,

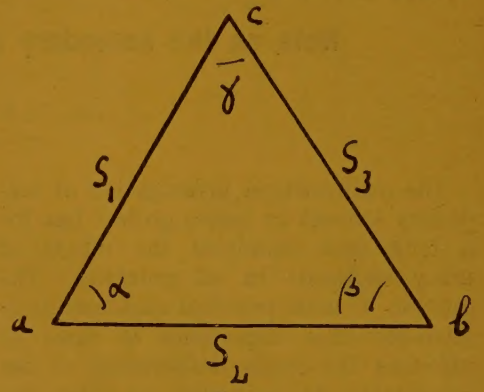


Fig. 1.

the lengths of the sides being l_1, l_2, l_3, the changes of angle $\delta\alpha$, $\delta\beta$, $\delta\gamma$ are expressed by the equations :

$$E\delta\alpha = (s_3 - s_2)\cot\beta + (s_3 - s_1)\cot\gamma;$$
$$E\delta\beta = (s_1 - s_3)\cot\gamma + (s_1 - s_2)\cot\alpha;$$
$$E\delta\gamma = (s_2 - s_1)\cot\alpha + (s_2 - s_3)\cot\beta.$$

It is obvious that $\delta\alpha + \delta\beta + \delta\gamma = 0$

Let any number of members $n1$, $n2$, nm meet at a joint n, and if τ_{n1} is the deflection angle of the member $n1$, the deflection angles of the following members will be :

$$\tau_{n2} = \tau_{n1} + \delta\alpha_1$$
$$\cdots$$
$$\tau_{nm} = \tau_{n1} + \sum_1^m \delta\alpha.$$

Considering the two ends of a member of a lattice girder, if τ_1 and τ_2 are the refer-

ence deflection angles of the ends reckoned positively in the same direction (for example, clockwise) the resulting moments can be expressed by the equations :

$$M_1 = \frac{2EI}{l}(2\tau_1 + \tau_2),$$

$$M_2 = \frac{2EI}{l}(2\tau_2 + \tau_1)$$

By putting $\Sigma M = 0$ for each joint, we obtain as many simultaneous linear equations as there are joints, the unknowns being the reference deflection angles τ_1. We can calculate the value of τ for each joint and the corresponding bending moments at the ends of every member.

The usual expression for bending

$$\sigma = \frac{Mv}{I}$$

allows us to calculate the secondary stresses resulting from these assumed conditions.

The choice of an arbitrary reference angle τ and the calculation of these τ in order of the changes of angle form the bases of Manderla's simplified method.

When using *Mohr's method* the *deflection angle of the joint* is taken as the original unknown quantity. This angle is that through which the whole rigid joint turns during the deformation of the girder.

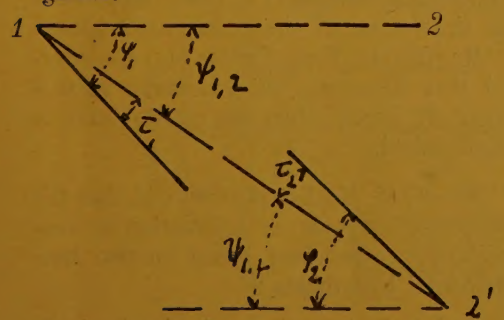


Fig. 2.

The member originally in position 1-2 occupies the mean position 1-2′ when deflected and has moved through the angle $\psi_{1,2}$. The rigid joints 1 and 2 have moved through angles φ_1 and φ_2 which are constant for each of the members of the same joint (the joint being supposed to be indeformable) forming the angles τ_1 and τ_2 which give rise to the secondary stresses.

$$\tau_1 = \varphi_1 - \psi_{1\cdot2}$$

$$\tau_2 = \varphi_2 - \psi_{1\cdot2}.$$

$$M_{1\cdot2} = \frac{2EI}{l}(2\tau_1 + \tau_2) = \frac{2EI}{l}(2\varphi_1 + \varphi_2 - 3\psi_{1\cdot2})$$

$$M_{2\cdot1} = \frac{2EI}{l}(2\tau_2 + \tau_1) = \frac{2EI}{l}(2\varphi_2 + \varphi_1 \quad 3\psi_{1\cdot2})$$

Manderla's method makes use of the angles τ themselves, while Mohr's method substitutes for them the deflection angles φ of the joints.

The angles ψ may be determined directly by numerical calculation or by graphics if a Williott displacement diagram for the deformation is drawn. The construction of this diagram avoids the lengthy calculations of $\delta\alpha$, which are necessary in the application of Manderla's method.

If the end of a member has moved through a normal distance λ measured on the diagram, the angle $\psi = \frac{\lambda}{l}$.

So far as physics are concerned, Mohr's method determines the angles φ, which best shew the deformation of the girder.

Mr. von Abo has applied in turn the different methods to a railway bridge having main girders of the Warren type with vertical members, and has come to the conclusions, corroborated by most engineers, that Mohr's method is the quickest and the most practical. It is essential to proceed methodically and to systematically solve the series of simultaneous linear equations for τ or φ.

Mr. von Abo's paper contains some very interesting diagrams showing the influence lines for secondary stresses at the ends of the various members of the girder. We show these in figures 3 to 13 which follow.

The *written discussion* which followed the publication of this paper produced an expression of opinion from many American engineers thoroughly conversant with this question. The majority were of the opinion that Mohr's method is the most practical for the calculation of secondary stresses.

Mr. O. H. Ammann thought that the calculation of these stresses was really of little importance and that it would not appreciably decrease the « factor of ignorance » in the design of a structure. There were too many other factors concerned. It was therefore hardly likely that these calculations would enter into general practice. Moreover, Mr. Ammann drew attention to the increased rigidity due to the gusset plates at the joints, and considered that in consequence the calculated stresses were not the true maximum stresses.

It was very difficult to introduce these factors into the calculations. Observations show that the maximum stresses at the joint of the member with the gussets may be obtained by multiplying the theoretical stress by the ratio of the free length between the gussets to the theoretical length of the member.

These maximum stresses were therefore greater than the theoretically calculated stresses. Mr. Ammann believed, that the increased strength of the girder due to the rigidity of the gusset plates more than compensated for any increase in secondary stresses.

Mr. Turner considered that too much importance had been attached to secondary stresses. The torsion of certain compression members and their deformation under eccentric loading give rise to greater variations in the stress than the rigidity of the joint. It was wrong to attribute certain failures of *badly constructed* bridges in Germany to secondary stresses.

The gusset plates were of greater assistance in increasing the rigidity of the girder and decreasing its sensibility to vibration under the effect of impact than they were undesirable owing to the secondary stresses they caused.

Mr. Charles Evan Fowler had carried out direct determination of stresses on railway arch bridges at Niagara. The determination of the maximum secondary stresses only interested engineers in allowing them to adopt a higher unit stress in the material. The ideal would be to utilise the metal up to its elastic limit, all the conditions of stress being known and evaluated.

In the case of the Niagara bridge, stresses of from 16 000 to 20 000 lb. per square inch exist (the elastic limit of the material being 30 000 lb.). The experiments carried out on this bridge have shown that the secondary stresses are not dangerous. It may be mentioned that no appreciable secondary stresses were found in the region of the large gusset plates.

Mr. Charles Evan Fowler also considered that there was an automatic adjustment of stress when the elastic limit is approached.

Mr. H. M. Mackay pointed out that the proposed methods of calculation of secondary stresses were based on two fundamental hypotheses :

1. that the joints were absolutely rigid;

Fig. 3.

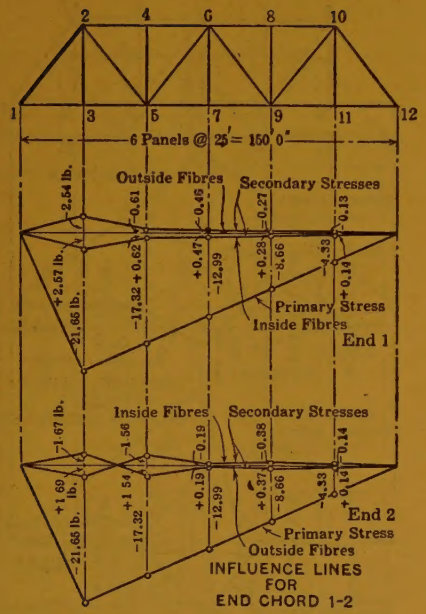
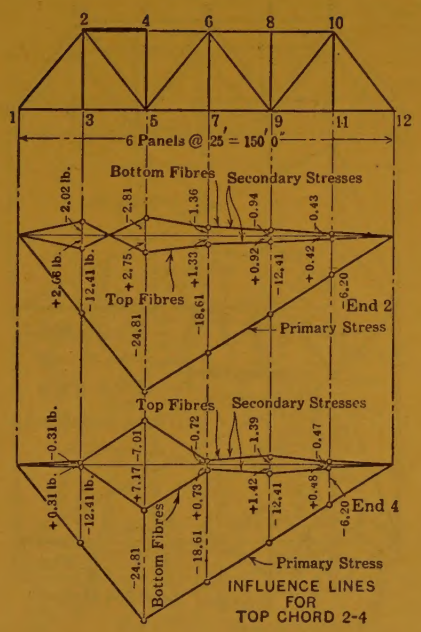


Fig. 4.



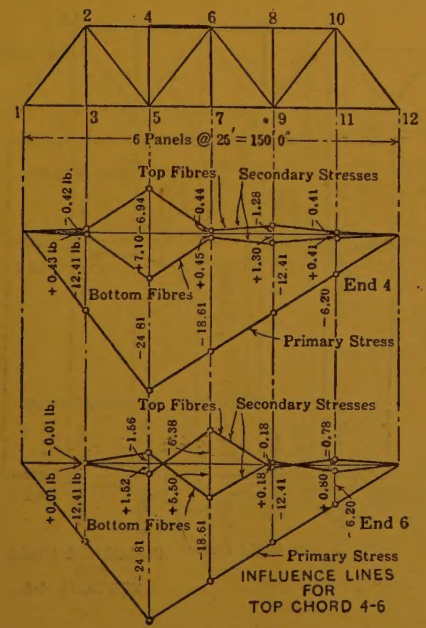


Fig. 5.

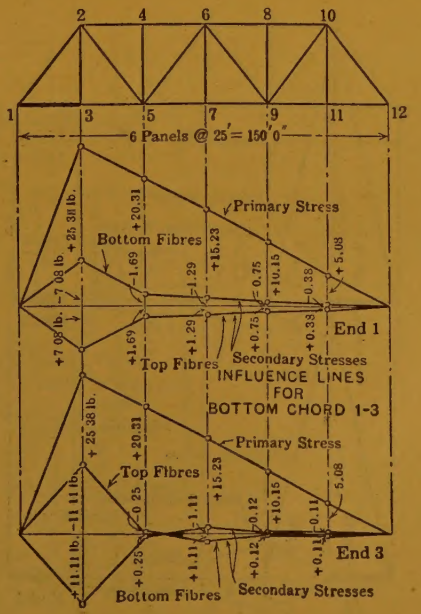


Fig. 6.

Fig. 7.

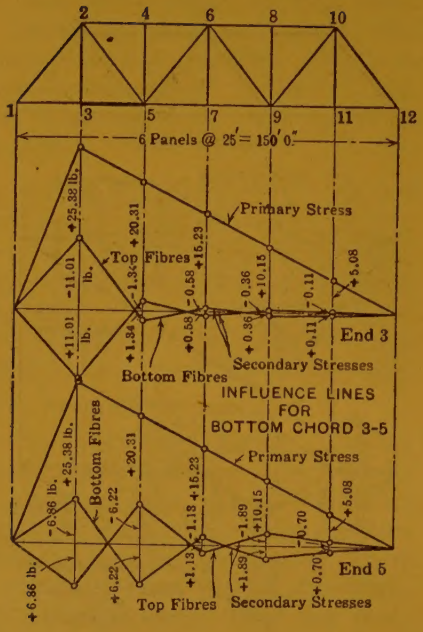
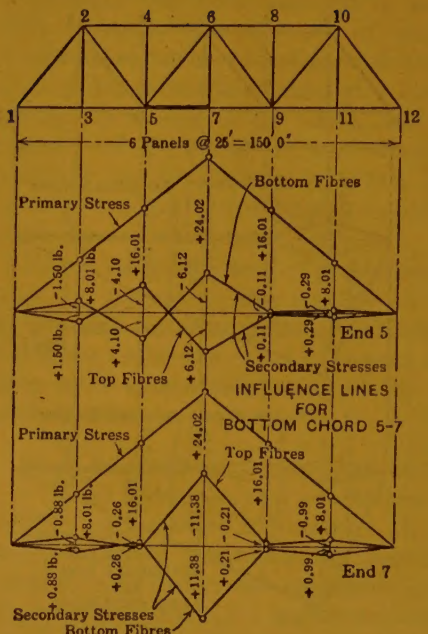


Fig. 8.



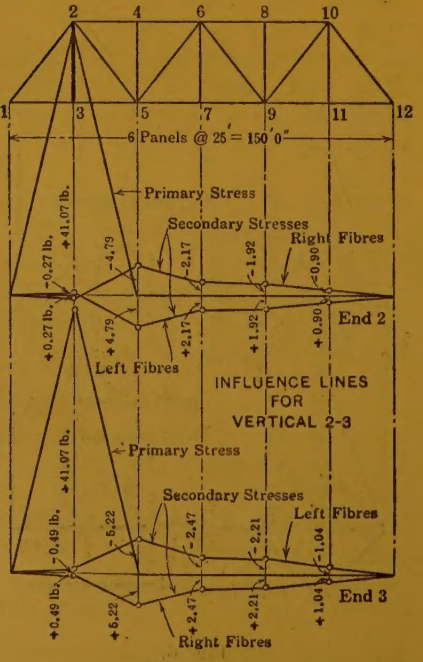


Fig. 9.

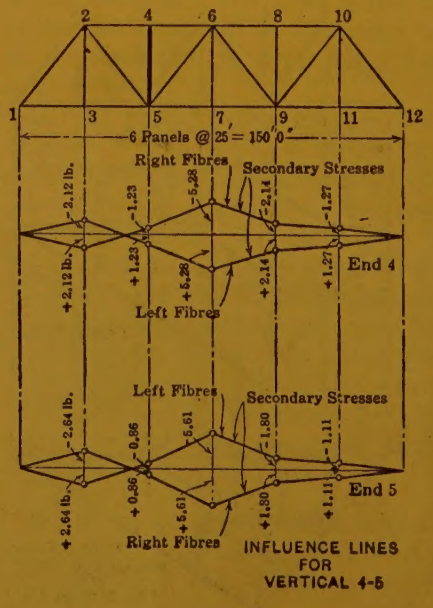


Fig. 10.

Fig. 11.

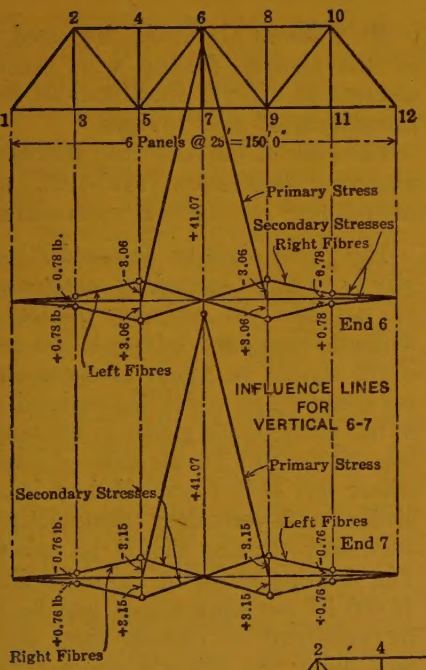
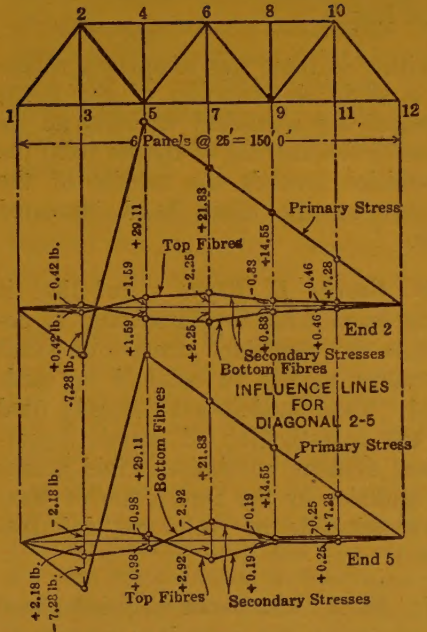


Fig. 12.



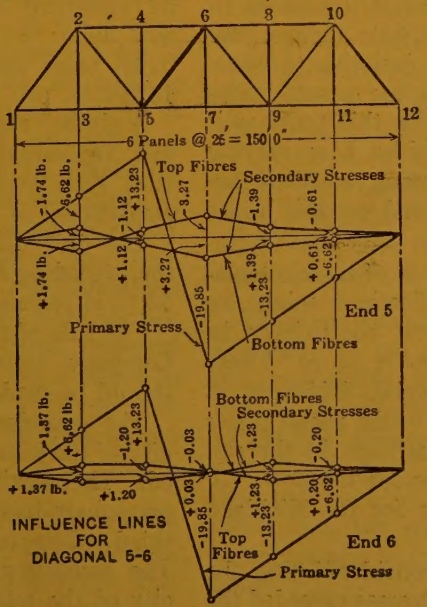


Fig. 13.

2. that the moment of inertia of the members remained constant along their whole length.

Neither of these was completely realised; a slight deformation of the joints decreased the secondary stresses; an increase of the moment of inertia from the extremities towards the middle of the member on the other hand increased them.

Mr. George E. Beggs stated that it would be possible experimentally to determine the secondary stresses on small laboratory models in celuloid or cardboard. These models were already used for the investigation of certain hyperstatical systems, continuous rigid arches and gantries by the application of Maxwell's principle of reciprocal deformation.

Mr. D. B. Steinmann had calculated and measured the secondary stresses in the arches of the Hell Gate Bridge. He considered that it was desirable to introduce the influence of the gusset plates in the calculations. For this purpose it was sufficient to consider I as varying along the member, being constant between the attachment, and variable under the gusset plates.

From the point of view of these calculations, the only modifications might be made in the numerical coefficients in the linear equations for τ or φ. There was therefore no special difficulty to overcome. The effect of the gusset plates had been taken into account in the calculation of the stresses for the new railway bridge at Niagara for the Michigan Central Railroad.

It was sometimes considered that the measured secondary stresses exceeded the calculated stresses. This was not the case if the calculations had been properly made.

Mr. Thomas C. Shedd discussed some of the clauses dealing with secondary stresses in official specifications for the calculation of steel bridges.

The rule of the American Railway Engineering Association (1920-1922), under the heading *secondary stresses* states :

The girder shall be designed in such a way as to avoid secondary stresses as far as possible. In ordinary lattice work without secondary triangulation no account should be taken of these stresses for members in which the ratio of width in the plane of the girder to the length is less that 1/10. In the case of members where this ratio is exceeded or in lattice girders with secondary triangulation, the secondary stresses should be calculated. The limiting stress in the metal fixed at 16 000 lb. per square inch may be raised by one-third to include this, provided that under the primary stress alone the metal shall not be stressed more than 16 000 lb. per square inch.

A similar rule is contained in the official regulations for the calculation of steel road bridges of the United States Department of Agriculture.

The only difference between this and the preceding rule is that the increase of 33 % is reduced to 30 %.

The rule of the New York Central Lines Company states that the calculations should be made in every case.

The secondary stresses due to the rigidity of the joint are always to be calculated. The maximum stress of 18 000 lb. per square inch may be increased by one-third when taking these into account. The secondary stresses due to eccentricity of the attachments of members and to transverse bending are also to be cal-

culated. No increase in the working stress is allowed in this case.

The Pennsylvania Railroad also requires that the secondary stresses of every kind shall be calculated.

It would appear from these American specifications that a good deal of importance was attached to secondary stresses. Actually the increase in the working stress was arbitrary; it was convenient to fix one maximum stress for the sum of the two stresses primary and secondary. Only in this way could the factor of safety be increased and be equal in all the members of the girder.

The calculation of secondary stresses in accordance with the methods given was not always entirely satisfactory, in that they did not take into account the variation in the moment of inertia. Large gusset plates at the ends might have considerable effect on the distribution of the stresses whilst inside the gusset plates themselves the distribution of the stresses was inknown. Certain engineers believed the ratio of the breadth of the member to its length to be taken into account as the principal factor. The question then arose what length was to be taken, the theoretical length between the joints or the free length between the gusset plates. In brief, whilst the problem had been very fully examined from a mathematical point of view, no light had been thrown upon a number of interesting technical points which would assist the designer to get out designs for the strongest and most durable structure for a given weight of metal.

Mr. Hardy Cross asked what immediate use could be made of the calculations of secondary stresses. It would in the first place be unlikely that anyone would modify the design of the girder in order to eliminate these stresses. On the other hand it was not to be recommended in the present state of our knowledge of the subject that we should design the members for the combined effects of primary and secondary stresses with an increased working stress. Mr. Hardy Cross considered :

a) That secondary stresses were of less relative importance for members in tension than for members in compression;

b) In compression members the secondary stresses at the extremities with opposite signs might be advantageous seeing that they would oppose deflection in any one direction and tend to form an intermediate point of contraflexure;

c) The secondary stresses were less important than the primary stresses, because of their localisation; this was especially the case when dealing with deflections in two planes perpendicular to each other;

d) At the present time we know very little of the influence of impact on the secondary stresses; it was possible from this point of view that gusset plates had the effect of diminishing the intensity of the vibration; they would therefore improve the conditions of the member from a dynamic point of view in spite of the fact that they gave rise to secondary stresses.

These considerations should not hinder the study of the problem, provided that the conclusions were merely concerned with the localisation of dangerous points without justifying an increase in the permissible working stress. Moreover, the true influence of secondary stresses on safety was far from being determined, having regard to the foregoing considerations.

Mr. Edward Godfrey was of the opi-

nion that in the first place one should only consider under the term of secondary stresses the stresses due to the rigidity of the joints. It would in the first place be interesting to know if the stresses were really of sufficient importance to justify the long and laborious calculations required to determine them. The writer was of the opinion that these stresses arising from deformation need not be considered. No bridge had failed on account of secondary stresses and no structure well designed and well constructed would fail through deformations of this kind. It must not be forgotten that steel is a ductile material and is not brittle like glass, and that it can in consequence undergo certain deformations which would be dangerous in the case of brittle material.

An engineer familiar with existing structures might recall a number of cases in which the theoretical stresses should lead to the failure of the structure. This proved that in many cases there was an actual adjustment of stresses which automatically produced a state of equilibrium, and this justified the secondary stresses being ignored. It was well to differentiate from the point of view of elasticity between stable equilibrium and unstable equilibrium (buckling) of a loaded strut, for example, or of the bending of a member subjected to repeated alternating loads in opposite directions. A bar of mild steel when tested can undergo, without cracking, bending round a very small diameter, while it would quickly break if bent in alternate directions.

It could therefore be assumed as regards the secondary stresses resulting from the rigidity of the joints — and this opinion was shared by Messrs Steinmann and Ammann — an actual readjustment of stresses occurred in practice.

In practice, imperfections in the construction and erection of structures caused higher secondary stresses than those with which we have been dealing. The malleability and the ductility of steel are essential properties which allow the metal to undergo the necessary deformation, and the same properties come into play as regards deformations caused by the rigidity of the joints. An important example of a structure in which numerous secondary stresses occur is the Forth Bridge. This structure owes its exceptional strength to the excellent design and notably to the rigid joints which give rise to secondary stresses.

Mr. F. P. Shearwood. — The object to attain was to construct a strong structure, as rigid and as durable as possible, with the minimum of material. The theoretical calculation of the stresses was necessary, not from an academic point of view, but in order to compare the calculated stresses in the structure with those in other structures which have proved to be satisfactory. The secondary stresses were attributed to the rigidity of the joints, and it was sometimes assumed that these would disappear if the structure were pin jointed. Actually these stresses occur in all lattice work structures. Unlike the primary stresses, they disappeared or were reduced by slight deformations of the metal. They could not therefore have any great influence on the safety of a structure.

The writer agreed on many points with Mr. Godfrey. A practical engineer knew that even in those structural steel works where the greatest care and accuracy was exercised, the metal was always subjected to stresses which were higher than those in normal service. It was rather strange therefore that steel, which was the most

ductile of metals, was the only one in which this property was deliberately neglected. All statically indeterminate stresses were avoided and no reliance was placed upon the continuity of the members. The determination of secondary stresses would be of interest if it were proved experimentally that they could have an appreciable effect on the stability of the structure. Actual results observed in practice shew that the features which gave rise to these stresses were rather a source of strength than of weakness.

Mr. Lawrence J. Waller. — The calculation of secondary stresses was based on three fundamental assumptions :

1. That the angles at the joints remained perfectly constant;
2. That the members were deformed elastically between the joints;
3. That the members had a constant moment of inertia throughout their whole length.

In practice the rigidity of the gusset plates was a variable factor. If some had moments of inertia comparable to the moment of inertia of the members, others had reduced moments of inertia. Moreover, the means of attachment — rivets — were not always perfectly rigid and a certain amount of movement might take place between the rivetted parts.

In any case these remarks shewed that it was misleading to treat this question with too great mathematical exactitude.

Owing to the variable inertia of the gusset plates, their action on the members was indefinite and could not be accurately determined by mathematical methods. Moreover, in practice, on large structures the moment of inertia was never constant throughout the whole length of the member.

The value of the stresses — primary and secondary — calculated by the ordinary methods was modified by the following circumstances :

1. Influence on the axial primary stress of the increase of section due to the gusset plates and attachments;
2. Influence on the bending stress arising from variations in the moments of inertia of the gusset plates;
3. Errors in workmanship, incorrect spacing of rivets, improper fitting of surfaces in contact under compression, slight errors in setting out, etc.;
4. Initial stress due to cold straightening the members;
5. Difference in temperature between various elements of the girder;
6. Bending between the joints due to the dead weight of the member;
7. Bending of the upright members due to the attachment of the floor members;
8. Torsion in the booms due to vertical rotation of the ends of members resulting from partial floor loading;
9. Longitudinal stresses due to friction in the expansion fittings;
10. Decrease in stress due to the action of the flooring as a member in tension;
11. Uneven distribution of the stresses throughout the rolled section.

It was only necessary to quote these causes of variation to illustrate the uselessness of carrying out with extreme accuracy calculations for secondary stresses, and consequently Mohr's semi-graphic method might be considered to be sufficiently accurate.

As regards the constancy of the angles at the joints, it might be mentioned, however, that the most recent Swiss experiments seemed to prove the validity of this assumption in the calculations of secondary stresses.

Mr. R. McC. Beanfield also drew attention to the causes of error existing in the calculation of secondary stresses from the assumptions that I remained constant and that the angles at the joints remained constant. This last assumption is largely invalidated by the movement at the rivetted joints and by the elasticity of the gusset plates.

The author believes that instead of making calculations of doubtful value, it would be much better to pay greater attention to the length of members during erection, by lengthening or shortening them according to the changes expected. This would get rid of most of the secondary stresses.

* * *

The summary of the written discussion on Mr. von Abo's paper, gives briefly all the arguments which have already been, and can be, put forward on the question of secondary stresses.

The diagrams (figs. 3 to 13) showing the influence lines for the stresses indicate their relative magnitudes in the example taken.

It may be noted that generally speaking the angular variations are proportional to the primary stresses. The secondary stresses are therefore proportional to the primary stresses.

If all the members of the lattice girder have the same moment of inertia and the same length, and if they are symmetrical, the equations show that the bending moments are proportional to $\frac{I}{l}$. On the other hand, if H is the depth of the symmetrical section the stresses are proportional to $M\frac{H}{I}$. Taking in account the ratio of M to $\frac{I}{l}$, it is to be concluded that the secondary stresses are proportional to H, the depth of the section in the plane of the girder and inversely proportional to L, the length of the member.

* * *

Particular stress is laid upon these conclusions in a report of the « American Railway Engineering Association »: *Iron and Steel Structures*, January 1914, as the result of experimental tests and calculations on lattice girder railway bridges.

A summary of these interesting conclusions is given below :

1. In a lattice girder, the deformations being proportional to the primary stresses, the secondary stresses are proportional to the primary stresses whatever the loading;

2. Other things being equal, in a symmetrical section the secondary stresses are proportional in the ratio $\frac{H}{L}$, H being the depth of the member in the plane of the girder, and L the length of the member.

Therefore, in two girders of the same type in which the length of the members and the moments of inertia are proportional, if in one the ratio of the depth of the members to their length is double that in the other, the secondary stresses will also be double in proportion;

3. The secondary stresses for any member depends upon the angular deformation of all the others, but mainly on the angular deformation of the adjacent triangles;

4. These conclusions enable the value of the secondary stresses in any given type of girder to be determined approximately beforehand when the ratio $\frac{H}{L}$ is known;

5. The stresses are lowest for members of constant dimensions. Sudden changes in the length and moment of inertia cause large variations in the secondary stresses;

6. Triangulation by equilateral triangles without vertical members produce the minimum stresses.

The stresses increase with triangulations of right-angle triangles, the more so as the ratio of the depth of the girder to the width of the panels is increased;

7. Intermediate uprights which serve to carry isolated loads (cross girders) produce important secondary stresses at their junction with the booms.

The best arrangement for these uprights is to incorporate them in the lattice work itself with a stress of equal intensity to that of the other parts and proportional to the loads carried;

8. The secondary stresses being proportional to the ratio $\frac{H}{L}$, they are particularly high in the secondary triangulation lattice work of the Baltimore and Petit types.

As regards the proportional numerical magnitude of the secondary stresses, this report also gives the following interesting information :

— It appears both from tests and calculations that the maximum secondary stresses do not exceed 40 °/₀ for a proportion $\frac{H}{L} \sim \frac{1}{10}$,

being therefore $4 \times \frac{\text{depth of the section}}{\text{length of the member}}$.

Without secondary lattice work, it is probable that these secondary stresses do not exceed 35 °/₀.

In through bridges the secondary stresses in the booms do not exceed $5 \times \frac{\text{depth}}{\text{length}}$ proportion.

The author of this article has carried out calculations of the secondary stresses in the case of a Warren girder with simple triangulation without uprights and having a span of 34.40 m. (112 ft. 10 5/16 in.) (four triangular panels of 8.60 m. [28 ft. 2 9/16 in.]) and of 4.75 m. (15 ft. 7 in.) high.

The results obtained confirm the American conclusions. The maximum secondary stresses occurred in the booms in the ratio of 15 to 16 % of the primary stresses, corresponding to ratios 3 to $4 \times \frac{H}{L}$.

As far as concerns the influence of the uprights and of the secondary members of the lattice work, we consider that their true function in the whole structure cannot be judged in accordance with the secondary stresses which they set up. Their most important purpose (beyond the subdivision of the panels) is to stiffen the girder and to considerably increase resistance to buckling of the members in compression.

* * *

We give below *the conclusion of Dupuy* (formerly « inspecteur général des ponts et chaussées » of France) as the result of calculations and trials made on the railway bridge at Cosne. He recommended the adoption of the type of girder which most nearly approached a *girder with a solid web* having a symmetrical section about the centre of gravity and if blank spaces were made *symmetrically* they should be arranged with reference to the axis of the girder. This led to the *exclusion of the N type girders* with simple triangulation and gave preference to the cross lattice girders with multiple lattice work.

* * *

To conclude this article, we may point out that in a recent publication of the Indian Government on the calculation of bridges for broad gauge railways (5 ft. 6 in.) we also find a clause dealing with secondary stresses. A distinction is made between deformation stresses due to the rigidity of the joints, and permanent secondary stresses, due to the eccentricity of the attachments, deflection under an isolated load, and deflection due to the weight of the members, and wind pressure. The former are called *deformation stresses;* the structure should be constructed in such a manner as to avoid redundant members and deformation stresses. In a well designed girder there is no need to concern oneself about these since they *tend to reduce themselves* (by deformation) and are not cumulative. One should, however, always take into account rigidity of the attachments in the calculation for the rivets.

The latter stresses are known as *secondary stresses;* they should be calculated, as these stresses are additional to the primary stresses. Unlike the deformation stresses, these are not *self relieving*, and do not tend to neutralise themselves.

R. Despretz.

[625 .144.4]

Mechanically operated tools used in permanent way maintenance,

By Dino LEVI de VEALI,
ENGINEER OF THE NORTH OF MILAN RAILWAY.

Figs. 1 to 6, pp. 293 to 295.

(*Rivista dei Trasporti.*)

For some considerable time there has been much discussion, together with many tests as to whether power-operated tools could be used with advantage in maintenance work on railways and tramways. It may seem strange that there should be such great difficulty in introducing into Europe methods that have been in use in America on a large scale for some time. Another interesting point is that the methods and tools used in permanent way construction and maintenance have lagged far behind the great technical and industrial progress of recent years, and are still practically the same as those in use when railways were first built.

The question is to-day one of the greatest importance, and in future will be even more so in view of the increasing difficulty experienced in recruiting the men required for the maintenance gangs. The hard work, the limited pay, the need to live away from large centres, as well as other reasons, tend to make the men prefer to work in factories.

The cost of upkeep as a whole, and in particular the cost of the renewal of the track, form an important part of the costs of operation, so that it is much to be desired that every effort should be made to effect the largest economies possible without affecting safety.

Consequently the use of power-driven tools seems at first sight an efficient remedy, either for making good any shortage of staff, or to make the work less trying, or to allow higher rates of pay in proportion to the fewer men needed, and to the specialised work done by them.

As we shall see, there are certain difficulties in the way of the introduction of these tools, and certain characteristics of our railways and methods of upkeep make it doubtful if any saving will be effected by their use.

It is none the less desirable that the tests should be continued to see if it is not possible to overcome the disadvantages already noted, by altering, if need be, the organisation of the maintenance gangs.

With these ideas before it, the North of Milan Railway decided to test such appliances on its lines. The results, as we shall see later, were unfavourable : this conclusion was only come to after having carefully considered if the plant could be kept fully employed, and also the many difficulties that would have arisen if the staff had been completely re-organised.

A brief summary of the results of the trials will be of interest.

The tests were carried out during the summer of 1925 when renewing the formation of the section, Morosolo to Gavirate, on the Varese-Laveno line. Although differing from the usual maintenance job, this work was very suitable to shew

in a comparatively short time the results to be expected from power driven equipment, and to give definite comparisons with manual labour. The experimental equipment was supplied by Messrs. Nicola Romeo & Co.

The tools were worked by compressed air supplied by a monobloc motor compressor (fig. 1). The principal features of this compressor are that the two centre motor cylinders and the two outside compressor cylinders are cast in a single block. The motor cylinders work on the 4-stroke cycle, the ignition being by high tension magneto. The compressor cylinders are single acting. There is no automatic regulator, any air in excess of the consumption being discharged by a valve on the reservoir which could be set to different pressures. At the normal speed of 1 400 revolutions, the unit compressed 2 000 litres (70 cubic feet) of free air per minute to a pressure of 8 atmospheres (118 lb. per square inch) when developing 15 H. P.

The reservoir was connected to the compressed air collector by flexible metallic piping, and was therefore separate from the machine.

The other features were :

Diameter of motor cylinder	100 mm. (3 15/16 inches).
— compressor cylinder	85 — (3 11/32 —).
Stroke.	130 — (5 1/8 —).
Theoretical volume of free air drawn in per minute.	2.070 m³ (73 cubic feet).
Maximum working pressure	8 atm. (118 lb. per sq. inch).
Dimensions :	
Height	1.200 m. (3 ft. 11 1/4 in.).
Width.	0.710 m. (2 ft. 3 15/16 in.).
Length	1.280 m. (4 ft. 2 3/8 in.).
Weight :	
Motor compressor without reservoir	400 kgr. (882 lb.).
Reservoir.	62 — (137 lb.).

The tools used were the following :

a) Four tampers made by the Ingersoll-Rand Co. (fig. 2);

b) Pneumatic drill for drilling sleepers (fig. 3);

c) Coach screw driver (fig. 4).

The monobloc compressor was too small for the whole of the tools to be used together : it was realised that eventually a larger compressor would be necessary.

Results obtained so far as quality of work is concerned. — So far as the drilling of the sleepers and the driving of the coach screws were concerned, there was not much to note. The drilling was perfectly regular, the treenails were driven to the bottom of the holes when the friction joint acted to prevent them from being driven any further. It would be an improvement if the drills were fitted with a small pipe in line with the drill point so that the shavings might be blown out of the holes.

The tamping, which is the characteristic feature of permanent way maintenance, was more interesting. The more or less great superiority of mechanical tamping has been often discussed, opinions being much divided. This division is probably due to the peculiarities of the tools used. During the test we found that as a result of the angle at

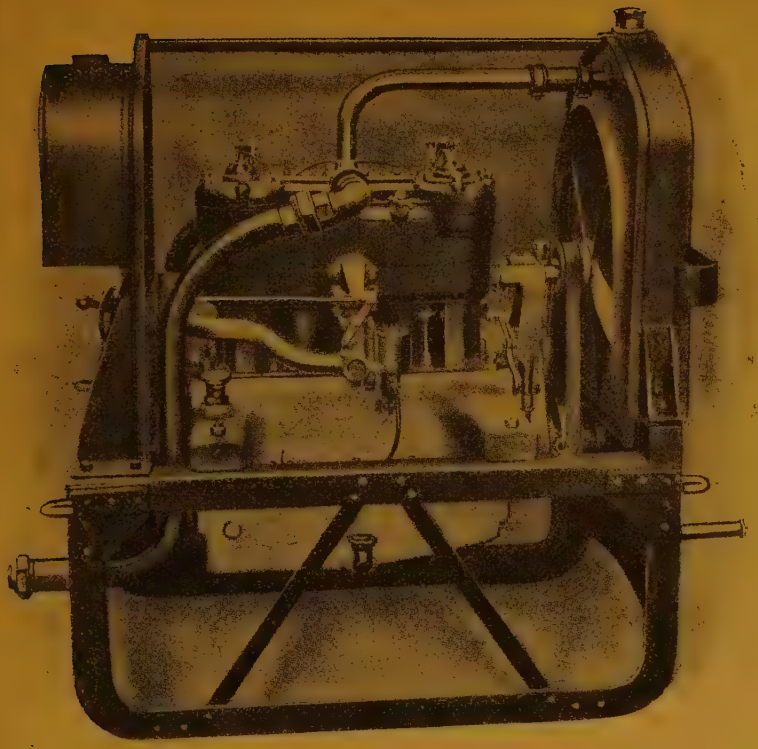


Fig. 1. — Monobloc motor compressor.

Fig. 2. — Mechanical tamper.

Fig. 4. — Coach screw driver.

Fig. 3. — Pneumatic drill.

which the tamps always work, although it may be lessened at the final stages of the work, the ballast immediately under the sleeper is not so well consolidated : but on the other hand the tool acts to a greater depth than the tamping pick. With hand tamping the consolidated layer was more compact on the surface, but was not so thick.

As a whole, the results, especially when the men had become conversant with the new method, were considered satisfactory, and in no way inferior to those obtained with hand tamping.

The notable advantages of mechanical tamping were the following :

1. Ability to properly tamp even the inside edges of joint sleepers, which is difficult to do by hand owing to the close spacing;

2. Ability to properly tamp crossing timbers, and in general at all places where hand tamping is difficult to do;

3. Less breaking up of the ballast, thanks to the simple vibrating motion by which it is driven under the sleepers;

4. No damage to the edges of the sleepers, which is unavoidable when tamping with the tamping pick.

Most of the men welcomed the air driven tools as they were less tired after using them, although at first the vibrations of the tampers were very trying.

Quantitative results and comparative costs. — The amount of work done hourly with the air driven tools was measured towards the end of the test, that is to say, after about 20 days' use, and was based on the average of a number of hours of effective work, the interruptions due to traffic requirements or other causes being excluded.

The drilling of sleepers and driving of coach screws was always done by two men, who worked in turn the drill and the driver. The two men were able to drive 100 or 120 coach screws per hour, although it is possible that with greater experience and better arrangements, the figure would be increased to 150 or 200. To be on the safe side, the figure of 120 was used in the calculations. By hand, with two men, it is difficult to get more than 25 to 30 coach screws driven an hour. Two men working continuously with the pneumatic tools were able therefore to do as much as 8 men working by hand, so that each pair of tools would save 2 men.

It is to be noted that owing to the compressor being too small, the tampers could not be used at the same time as the drills and coach screw drivers.

In considering the relative savings due to the compressor tested (based on the cost of hire) the saving due to the tampers only was taken into account.

So far as tamping is concerned, a gang of 5 men was used, four working the pneumatic tampers and the fifth looking after the piping. The latter appeared to be required to prevent the flexible pipes to the last section becoming twisted and thereby broken. In one hour, when not interrupted, the gang tamped 60 sleepers : it is probable that when the men became more expert the number would be increased to 70 or 80.

A gang of 8 men working with tamping pick are only just able to tamp 25 to 30 sleepers an hour. Each group of 4 tamps at work effects a saving of 12 to 13 men.

In making a comparison of cost, two cases are considered.

In the first, the actual cost of the work is considered, that is to say, in the cost is included the 75 lires a day the contractor charged for the hire of the compressor and the tools; in the second, the purchase of a larger compressor is taken into account. As stated above, the compressor was too small for the tampers, drills and coach screw drivers to be kept at work together. In the first case, the economy of labour due to the use of the

Fig. 5.

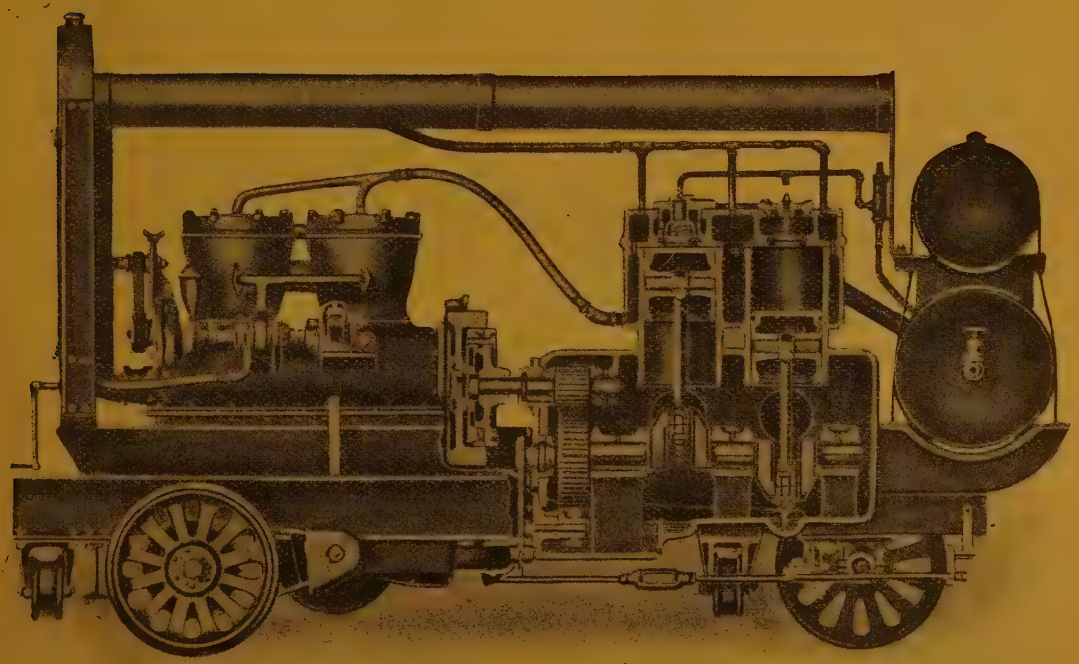


Fig. 6.

pneumatic tampers has alone been taken into account. The calculation therefore has been based on an eight-hour day, and the wages of the men at 24 lires per day.

First case.

Saving of labour :		
12 men at 24 lires		288 lires.
Extra expenditure :		
Hire of compressor and tools	75 lires.	
Wages and travelling allowance of fitter	50 —	
Petrol and oil	115 —	
		240 —
Saving per day		48 lires.

Second case.

In the second case, supposing a compressor has been purchased, the corresponding daily cost had to be determined. The following costs were taken into account :

a) fixed charges;

b) costs only incurred when the plant was working.

Under *a*) amortization of the capital and, for the sake of simplicity, the maintenance, expressed as rather a low per cent are included, and under *b*) the charges due to the driver's wages and to fuel cost.

The total amount of the first items has been divided up in accordance with the number of days allotted each year to permanent way renewal, which under most favourable conditions cannot be taken at more than 150.

The compressor in view (Ingersoll, fig. 6) was large enough for all the tampers to be worked at the same time as the drills and coach screw drivers : its price, piping and spares included, would be about 75 000 lires. The annual charge for amortization at 10 % was therefore 7 500 lires, and for repairs taken at 6 %, 4 500 lires. The total, 12 000 lires, divided by the 150 working days, gives a daily cost of 80 lires. We get therefore :

Saving in labour :		
For the tampers, 12 men at 24 lires	288 lires.	
For the other tools, 6 men at 24 lires	144 —	
		432 lires.
Additional charges :		
Amortization and upkeep, daily	80 lires.	
Driver of compressor	30 —	
Petrol and oil	150 —	
		260 —
Saving per day		172 lires.

The above figures are those that would be realised in practice if all the tools could be used continuously throughout the 8 hours of work each day. This is, however, far from the case under the conditions under which relaying and re-

pairs to the formation are usually carried out.

It must be explained that during the tests, the work was arranged, as far as possible, to reproduce all the features encountered during maintenance work. The renewall of the road bed had been completed previously for some considerable distance.

At night, during the intervals between trains, the old worn 9 m. (29 ft. 6 in.) rails were removed, and new 12 m. (39 ft. 4 in.) rails substituted, only the joint sleepers being moved. In this way the trains could be allowed to run though at reduced speed. During the day, the gangs, between the passage of trains, completed the work by taking out defective sleepers and renewing them, by relaying all the sleepers, screwing in the coach screws, straightening and levelling the track, and finally tamping the sleepers. Naturally, during each interval, the necessary steps were taken to clean the ballast throughout the length which it was thought could be completed before the next train was due. During a large part of the time taken for the removal and replacement of the sleepers, for levelling up and straightening the track, the pneumatic tampers could not be used.

Even with the best arrangement of duties, it was difficult to keep the compressor at work more than 6 hours a day.

In addition to those interruptions caused by service requirements, there were many others, though of short duration, due to accidents, such as damaged tools, leaks or defects at the pipe joints, breakdown of the motor, etc. During many of these interruptions, as well as when the men were moving from one part of the job to another, and during relief, the motor was left running, whereby there was some waste of fuel.

These disadvantages, which are always met with during permanent way work, seriously lessen the results shewn in the comparison of costs, and also absorb all the savings shewn.

Putting aside these last considerations, that is, presuming that it would be practicable to work the plant continuously, and thereby get the savings suggested in case 2, it will be interesting to look at the saving from another and more practical point of view. We shall then see more clearly the difficulties in the way of using these tools and, furthermore, in the way of organising the work as it ought to be done.

When working out the second example, we arrived at the daily saving in lires that would be given with a compressor large enough to enable all the equipment, 4 tampers, 1 drill, and 1 coach screw driver, to be used at the same time. This supposition, whilst it is very important in the case considered, is of little importance when it is a question of deciding if the same method is to be used in the ordinary maintenance of a whole system or a line.

The points to be settled in this case are :

a) The proportional saving in men, or, in other terms, the reduction as a per cent in the staff employed to be obtained by a full use of these tools;

b) The saving in per cent on the costs of labour for ordinary renewal to be obtained by a complete use of this method.

For this to be done, the work of ordinary renewal must be analysed, and the time spent by the gangs on this work determined. Below are given figures which, whilst of no absolute value, are quoted only as average values or as indices, adopted to facilitate the calculations. We may take as the basis of comparison that one man in an hour can lay half a sleeper, that is to say can do all the work involved in renewing a length of line equal to half of the mean distance between the centre of two sleepers, in-

cluding removal and replacement of fittings, transport, replacement of sleepers, drilling of sleepers and driving coach screws, levelling up, straightening and tamping the ballast.

An investigation into the work done over long periods reveals the fact that a platelayer attached to a maintenance gang can properly inspect and keep in order daily three metres (3 1/4 yards) of line in an average state of repair. This figure corresponds to about 0.40 m. (15 3/4 inches) of work per hour, or, on an average, to a degree of activity represented by 0.5 sleeper-hour.

Let us suppose, although the figure is on the high side, that, when taking up or when replacing the sleeper, or correcting the gauge, replacing coach screws, bearing plates, or rails, 4 coach screws are removed and replaced per sleeper before redrilling, and, in the case of sleepers at points and crossings, a mean of 5, supposing half the bearing plates to have three holes and half two.

In an hour the platelayer should carry out :

the drilling and screwing in of 2 coach screws;

the tamping of half a sleeper;

the other work necessary to complete 0.40 m. (15 3/4 inches) of track.

So far as drilling and driving the coach screws is concerned, we have already seen that two men working together can put in place 28 coach screws an hour, the loss of time due to travelling, gauging coach screws, etc., being taken into account. We can then accept, as an approximation, an hourly rate of 14 coach screws per man, which is equivalent to the man driving a coach screw in about 4 minutes.

We have also seen that 8 men can tamp 30 sleepers an hour : to tamp one the workman takes about 15 minutes, including all lost time.

The hour in which the platelayer can completely go over a length of track equal to half the distance between centres of sleepers can therefore be divided up as follows :

A.	Drilling of sleepers and driving of coach screws.	8 minutes.
	Tamping the sleepers . .	8 —
B	— Other duties	44 —

In other words, the operations B, to be done by hand, required 88 minutes work per sleeper.

Now let us consider the question of tamping. We have seen that with pneumatic tampers the tamping of about 70 sleepers can be completed in about an hour. For the operation B for these sleepers, $88 \times 70 = 6\,160$ minutes are necessary, or, in round figures, 100 hours of manual labour. As a matter of fact, under the conditions given, the drills and coach screw drivers were unable to keep pace with the tamps: they should be able to deal with 280 coach screws per hour, whereas they have been taken as able to handle 200. This deficiency would only slightly modify the results, which are only approximations, so can be neglected, and we may take it that in an hour's work the whole of the power driven tools would be able to do the operations A relatively to 100 hours of operations B; in other words, for each compressor used, a gang of 100 men would be required to prepare and complete the work.

We can now ascertain the percentage reduction of staff obtained by the power driven tools needed to carry out operation A :

Manually :	
Men required to do the work given by the power driven tools	25
Men required to carry out the other operation *B*.	100
Total. .	125
Mechanically :	
Men working tools	7
Men required to carry out the other operation *B*.	100
Total. . .	107

The saving as a per cent is therefore about 15. Let us now see what the real saving is :

Manual labour :	
125 men at 24 lires	3 000 lires.
Mechanical work :	
107 men at 24 lires	2 568 lires.
Amortization and repairs . . .	80 —
Driver	30 —
Petrol and oil	150 —
Total. . .	2 828 lires.

The saving obtained is therefore 6 % on the labour costs.

The values given vary naturally from one line to another, and are closely related to the condition of the road-bed. In any case, it is as well to note that they are close enough to average values, and consequently a general conclusion can be based upon them.

As has been shewn, the saving is far from great, even under the hypothetical conditions of continuous output during the whole 8 hours' day. On this subject it should be remarked that, even if the plant were so perfected that the interruptions to continuous work were remedied, and the working of the line was so organised that the work could be carried on without interruption, each gang would have to be increased to 110 to get the desired results. This number might be reduced when use is made of machines to break up the ballast; the output of such tools is, however, doubtful.

Carrying out the repairs with gangs of this size completely upsets the organisation of the service. To-day the repairs are done by gangs of 10 to 12 men working over sections of 6 to 7 km. (3 1/2 to 4 1/2 miles). If reduced by 15 % to get the necessary economy, the number would be 8 or 10. For each compressor group, it would be necessary to collect 10 to 12 gangs.

Once this regrouping was effected, the constant and efficient inspection of its section by each gang would cease : the ganger would no longer be responsible, and the means of quickly carrying out any emergency repair would no longer be available. In fact, all the disadvantages resulting from the repair of the line by divisions of 50 km. (30 miles) instead of sections of 6 or 7 km. (3 1/2 to 4 1/2 miles) would be felt.

It would still be necessary to carry out all secondary duties, such as cutting hedges, cleaning out ditches, weeding, repairs in stations, now done by the same gangs. The regrouping therefore would only be possible during the time reserved for renewal, after which the old gangs would be re-formed. The result would be that the platelayers would retain their present houses so that it would be difficult, and involve much lost time, to convey them from one end of a section to another.

From these facts and reasons the conclusions readily appear : they are as follows :

1. The saving on labour costs, obtained by means of power-driven tools, is doubtful and possibly nil, both because of the defects in the tools and their cost, and because of the organisation of the work and the requirements of the operating department;

2. Even if the ideal conditions of continuous working could be realised, the saving obtainable (6 %) is not enough to justify a reorganisation of the repair work, which would give rise to too many difficulties of other kinds.

These conclusions are only applicable to renewal work; when there is question of laying new lines or of general improvement work, the power driven tools can be made use of with much profit.

The review of the question given above is confirmed in an authoritative manner by the discussions that took place during

the 10th Session of the International Railway Congress at London, June 1925. The first section (Way and Works), in general session, adopted the following conclusions (Question I : « Upkeep of the Track », article 8) :

« The obstacles encountered in Europe as regards the development of mechanical processes in routine maintenance work are generally of an economic nature : high cost of the mechanical equipment, inadequate daily output of the equipment resulting from the difficulty of co-ordinating mechanical work and manual labour on a given piece of work. These obstacles, however, are not met with in America, where considerable economies are effected by the use of not only small appliances such as tamping machines, but also of much more powerful machines, such as machines for screening ballast, liveling the track, etc.»

[625 .13 (.73)]

Great Northern builds longest railway tunnel in America.

Figs. 1 to 6, pp. 302 to 308.

(From *Railway Age.*)

The Great Northern is constructing a single track tunnel 7.78 miles long through the Cascade range between Berne and Scenic, Wash., which, when completed, will be the longest railway tunnel in America. The new line, of which it is a part, will lower the summit elevation 506 feet, will save 7.67 miles of distance and eliminate 6.04 miles of snow sheds, and 1941 degrees of central angle, besides reducing the maximum curvature from 10 to 6 degrees. Within the limits of the work the maximum gradient eastbound is reduced from 2.2 to 1.565 %. As a part of the project the present electrification through the existing tunnel will be extended to Wenatchee on the east and to Skykomish on the west, a total distance of 80 miles, and, later 23 miles further west to Goldbar, 29 miles east of Everett.

From an economic standpoint the construction of the tunnel can be fully justified by the elimination of the snow sheds on the old line with their high maintenance cost and constant fire risk. The snowfall in the Cascades is very heavy and at the point where the railroad crosses the mountains, amounts to 56 feet annually, which packs down to about 16 feet on the level. As a result of this extraordinary volume of snow, combined with the steep slopes of the mountains, snow slides are frequent and troublesome. Occasionally, they destroy sections of the snow sheds and interrupt traffic until the track can be cleared of the snow and debris. In addition to the elimination of the snow sheds, the railroad will receive the benefits of the decreased rise and fall, the shorter distance, the better alinement and the improved traction resulting from the electrification of the mountain territory, and these taken together are in themselves sufficient to justify the project.

The outstanding feature of the construction is the rapid progress that has been made to date (December 1926), as the world's record for advancing has been broken in four consecutive months, in three of which the advance was greater than in the preceding month. As a result, on 27 November the contractor was ahead of schedule by the amounts shown later. In August the pioneer tunnel at Scenic, 8 feet by 9 feet in section, was carried forward 937 feet; in September the 10-foot by 10-foot center heading at Berne was driven 954 feet, but counting 1 October, as all records of tunnel advance are based on 31 days, this was increased to 984 feet; and in October, a still higher record of 1 157 feet was made in the pioneer heading at Scenic. In November an advance of 952 feet was made in the pioneer tunnel at Scenic and 938 feet in the center heading at Berne, and while lower than the September and October records in this tunnel both are greater than any known record on other tunnel work. On 1 December, the total primary penetration was 17 257 feet or 42½ % of the total length of the tunnel.

Recently the Great Northern has been rearranging its engine districts and gradually increasing the distance between engine terminals. As a result of this policy, the rearrangement between Spokane and Everett eliminated one district, so that freight trains now run from Spokane to Wenatchee, 174 miles, and from Wenatchee to Everett, 132 miles. The electrification, which is a part of the tunnel project, will extend over the heaviest of the mountain grades, and the later extension to Goldbar will cover practically the entire mountain territory through the Cascade range. When electrified service begins, it is expected there will be a material reduction in the time required to move trains between Skykomish and Wenatchee.

At present, the existing line passes through a tunnel 2.63 miles long and it is electrified to reduce the trouble experienced from locomotive gases. Under the present system of steam operation heavy pusher service is required. The speed is slow and much time is lost in switching the pushers in and out of trains, while stops must also be made at Tye and Cascade, the portals of the existing tunnel, to attach and cut off the electric locomotives which pull the trains through the tunnel.

Under electric operation over the extended mileage, which will cover all of the heavy mountain grades, with the reduced grades and distance and the higher speeds of the trains while in motion, the movement of freight through the mountains will consume less time. Again, with the improved traction the number of trains will be less; operating costs should be reduced materially; and a substantial amount of power and car equipment will be released for other service. A detailed description of the electrification and the electric locomotives appeared in the *Railway Age* of 27 November.

The old line.

Westbound from Wenatchee, the line ascends the valley of the Wenatchee river and from Leavenworth to Chiwaukum it lies in Tumwater canyon, entering Nason Creek canyon at the latter point which canyon it follows to its upper end at Cascade, the east portal of the tunnel. The line emerges from the tunnel at Tye and enters the canyon of the Tye river which it follows to Skykomish, 21 miles. Between Tye and Scenic, 9 miles, the line descends 1 045 feet, and about two-thirds of the distance is through snow sheds. At Martins Creek, the line passes through a tunnel, both portals of which face east. When the new tunnel is put in operation, the existing tunnel 2.63 miles long, between Cascade and Tye and all of the old line between Scenic and Berne about two miles below Cascade, a total distance of 17.67 miles, will be abandoned.

It was stated at the beginning of this article that the new Cascade tunnel will

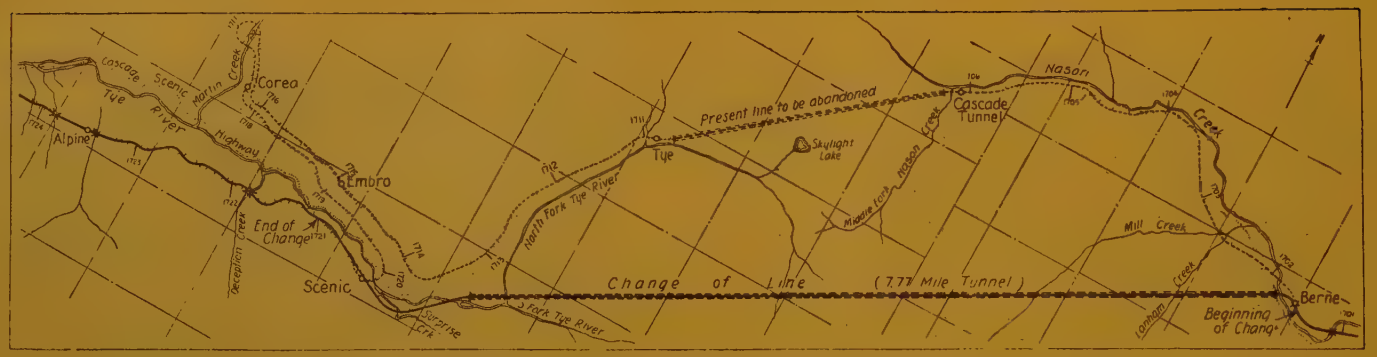


Fig. 1. — Map of the present and new locations between Berne and Scenic.

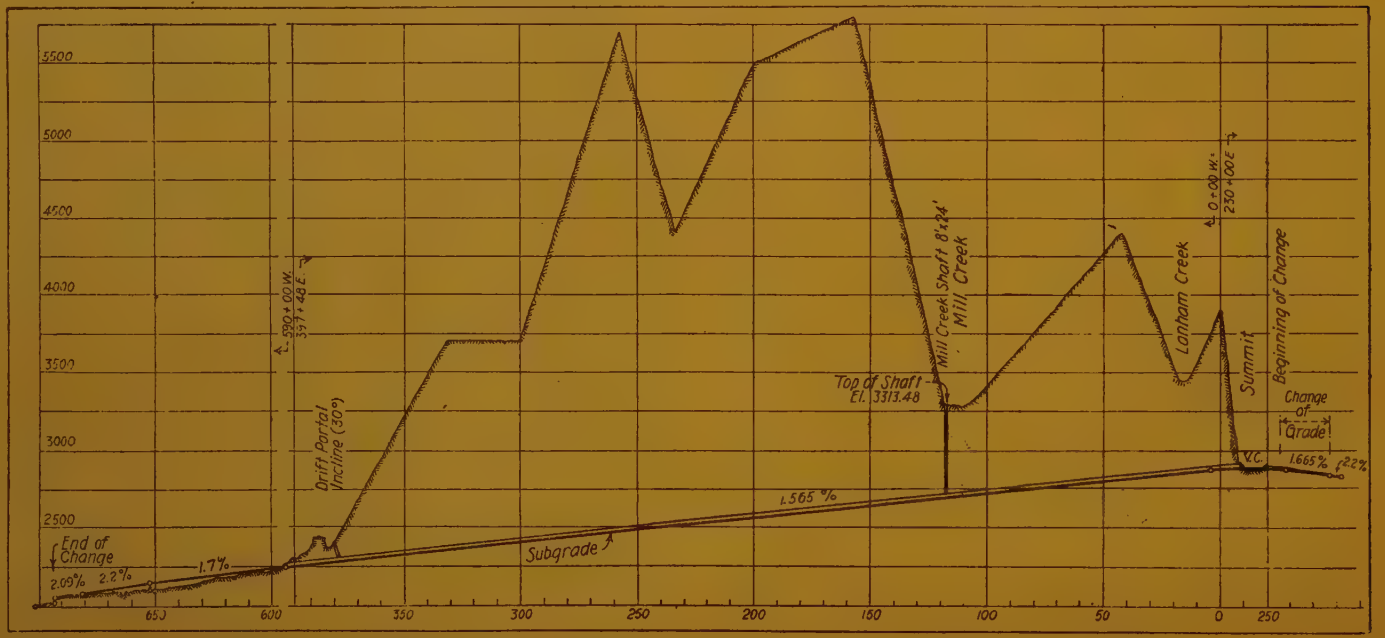


Fig. 2. — A profile of the tunnel showing the location of the Mill Creek shaft and the Tye River.

be the longest one in America; it will also be the fifth longest in the world. A comparison with some of the longer tunnels of the world follows:

NAME.	Location.	Length (feet).	(feet). Width	Height (feet).	Built.
Simplon I.	Switzerland	65 734	16.4	18	1895-1906
Simplon II	Switzerland	65 734	16.4	18	1921
St. Gothard.	Switzerland	49 212	26.2	24.6	1872-1882
Loetschberg	Switzerland	47 685	30	20	1906-1913
New Cascade	Washington	41 142	16	24	1925-
Moffat.	Colorado	32 160	16	24	1923-
Rogers Pass	British Columbia. . .	25 900	26.5	21.75	1913-1915
Hoosac	Massachusetts	23 175	25	20	1854-1876

How the work is organised.

Between Scenic and Mill Creek a pioneer tunnel, 8 feet by 9 feet in section, is being driven parallel to and 66 feet south of the main tunnel, with cross cuts at intervals of about 1 500 feet. From these cross cuts a center heading, 10 feet by 10 feet, is advanced on the line of the main tunnel, and this is being followed by enlargement to full tunnel section from the west portal and from the foot of the Mill Creek shaft.

Mill creek flows through a deep and well defined canyon about 2.4 miles west of Berne, and here a point was selected to sink a shaft 659 feet deep, so that additional faces can be worked both east

Fig. 3. — The head frame at the Mill Creek shaft ready to move into permanent position.

and west from the foot of the shaft. From Mill Creek a pioneer tunnel is being driven west, with cross cuts at intervals of 1 500 feet, to meet the pioneer advancing from the west portal, while a full tunnel section is also being driven west from the shaft. A center heading is being driven east from the foot of the shaft, for since the center headings between Mill Creek and Berne can be holed through in a comparatively short time, there is no advantage in driving a pioneer tunnel between these points. When the center headings from Berne and Mill Creek meet, enlargement will take place from the east portal.

It was desired to begin work on the center heading at Berne before a steam shovel was available to make the approach cut, and, as the topography was favorable, an adit was drifted in from the north until it met the line of the main tunnel about 200 feet west of the portal, and from this point a center heading was started. After the approach cut was completed a top heading was drifted on the line of the main tunnel and the bench removed as far as the junction with the adit, but no further work will be done towards the enlargement at this end until the center heading from Berne meets the center heading which is being advanced east from Mill Creek.

At the Scenic end, the first 800 feet were driven through soft material, consisting of clay in which large boulders were embedded and gravel streaks. This material was very wet, although the amount of water draining into the tunnel is comparatively small. At one point the bores crossed a seam of clay and gravel about 140 feet wide, which had the consistency of sloppy concrete, and gave considerable trouble both at the face and in the placing of the timbers. Because the general character of the ground was known, it was deemed advisable to start the center heading in the rock west of the first cross cut, and the top center heading method was followed from the portal to this cross cut. The section was then enlarged to a full top heading and behind this the bench was removed to give the full 18 feet by 26 feet tunnel section.

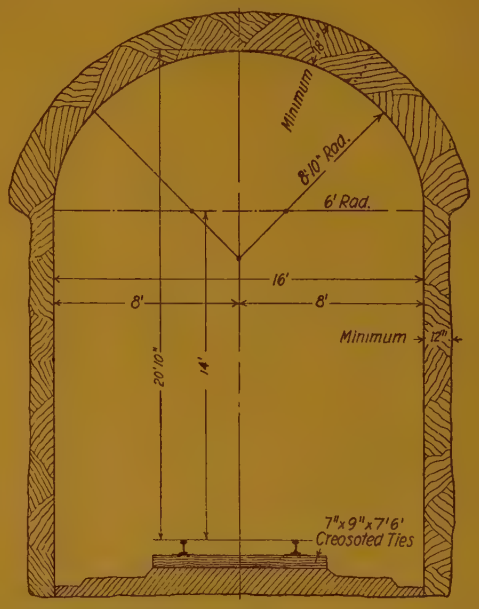


Fig. 4. — The standard tunnel section.

At the west end the tunnel emerges into the valley of the Tye river, which follows a tortuous course at this point and falls sharply in a series of cascades, so that while the stream is below the track level at the portal, it crosses the tunnel about 60 feet overhead at a point 2 000 feet farther east. About 225 feet east of the portal, a creek tributary to the Tye crosses 14 feet over the main tunnel and to avoid danger from flooding, a wooden flume was constructed to carry the water over both the main and pioneer tunnels.

Under the present channel of the Tye river a large crack was encountered which gave every evidence of being the result of an ancient slip. The rock seams were displaced both vertically and horizontally and the gap was filled with debris, consisting of clay, boulders, and logs, such as might be expected in a slide.

Presumably the Tye river changed its course when the slip occurred, as the clay pocket already mentioned appears to be in the ancient bed of a stream, because it contained boulders, tree trunks, and other drift.

The amount of timbering and other safety precautions necessary while passing through the soft ground made it certain that progress would be very slow until the headings reached solid rock. For this reason, an incline was drifted down from a point immediately east of the Tye river to the line of the pioneer tunnel, which was advanced from the foot of the incline 3 387 feet, while an advance of 2 316 feet was made from the portal. By this means the entire pioneer tunnel progressed 5 703 feet by the time the soft ground was passed, and four working faces for the center heading of the main tunnel were available through cross cuts.

The center headings at Scenic were started at the first and second cross cuts, and as soon as the heading between the first and second cross cuts was drifted through, a top heading was started east from the first cross cut. It had been intended, if possible, to enlarge the section from cross cut No. 1 by the ring method, but the coarse granite encountered in this section is highly impregnated with lime, which slacks after a few day's exposure to the air and the rock spalls badly, so that the ring method was not considered safe. The top heading permits the roof to be timbered before the benches are removed, so the men can work with safety. East of cross cut No. 2, the rock structure so far encountered will permit ring shooting, although there are occasional narrow seams where timbering will be necessary.

Character of rock.

The first rock encountered was a coarse granite which is very hard, but, as mentioned, is impregnated with lime and slacks upon exposure. It was difficult to drill and broke quite large, so that mucking operations were slow. After passing cross cut No. 2 the rock is an altered granite showing a much finer grain with occasional streaks of schistose structure, and, except for an occasional pocket containing mica, it drills easily and breaks in a satisfactory manner. At intervals, water under pressure has been encountered in the pioneer tunnel at Scenic and this, while small in quantity, has hampered blasting, so that several times it has been necessary to drill relief holes before the main holes could be charged for blasting. At one place the pressure was so great that the water jetted six feet from the holes. Peculiarly enough, however, the quantity of water has been comparatively small, and water under pressure does not occur in the center heading of the main tunnel, apparently, because the pioneer tunnel, which is in advance, affords the necessary relief to the pressure.

At the Berne end the first rock encountered was a coarse granite, similar to that at Scenic, which was hard to drill and broke large enough to hamper mucking. Timbering was required here also because of the tendency to slack. About 1 000 feet from the portal a finer granite, similar to that beyond the second cross cut at Scenic, was encountered, and only a small amount of timbering has been required in this formation. This rock is very dry and water has given no trouble at the Berne end.

The foot of the Mill Creek shaft penetrated a large block of graphite, and the graphite was quite widely dispersed in the rock surrounding the main deposit, so that solid timbering was necessary in the enlarged chamber at the foot of the shaft. The full tunnel section, 18 feet by 26 feet, was taken out for a distance of 100 feet, in both directions from the shaft to give working room for the disposal of the muck, and the full section will be continued west to permit the pioneer to advance more rapidly and make additional faces available, but the center

heading is continued east from the enlarged disposal station.

Disposal pockets were constructed below grade and gates installed for the purpose of loading the hoisting buckets. The muck is loaded in cars of 50 cubic feet capacity at the face of the center heading to the east, and in 6-yard cars at the face of the enlargement to the west. It is then hauled to the foot of the shaft by electric locomotives and dumped into the pockets. These pockets discharge into hoisting buckets to be lifted to the surface, where the material is emptied into storage bins from which it runs into the cars to be taken to the dump.

To permit an early start in sinking the Mill Creek shaft and avoid the delay which would have resulted from the construction of a permanent head frame, a small temporary head frame was erected so that the sinking operation could be carried on while the permanent head frame was under construction. This was built in two sections, the pockets being built in the position they were to occupy, and the head frame proper about 15 feet back of its final position to clear the temporary structure. When finished, the head frame was lubricated on its skids by forcing hard grease over the skids and onto the sills through pipes to which compressed air under 100 lb. pressure was applied. The head frame was then skidded forward by means of five 15-ton jacks, placed in exact position, and the shaft timbering was prepared with guides to match the guides fitted in the head frame. The time required to remove the temporary structure and place the permanent head frame was nine hours. The head frame is 80 feet high and is substantially built of heavy timbers. Each of the five long braces at the rear is a single timber in the form of a truncated square pyramid, 84 feet long, and having end dimensions of 14 inches by 14 inches and 18 inches by 18 inches, while the vertical members range from 12 inches by 12 inches to 16 inches by 16 inches.

The method of driving the full tunnel section in both directions from the foot of the shaft was to advance a top heading and follow this with the removal of the bench. Before there was room to operate a mucking machine and the model 20 Marion shovel, a small Hoe scraper with a tugger hoist was used to slush the muck into the pockets. As soon as there was sufficient room for the shaft disposal station, the center heading was started east and is progressing rapidly. West of the shaft the full tunnel section is being advanced by means of a top heading and removal of the bench. About 20 October, work was started on the pioneer tunnel and this will be carried forward as rapidly as possible. The pioneer tunnel will advance more rapidly than the main tunnel, so that a center heading can be started at the second cross cut west of the shaft and the enlargement in that direction carried on continuously.

Tunnel is comparatively dry.

As has been mentioned, water has given comparatively little trouble to date. The maximum amounts which have been encountered are as follows :

	Gallons per minute.
Scenic pioneer	125
Scenic incline	215
Scenic main tunnel	10
Mill Creek shaft	185
Berne center heading	2

Between 1 June and 1 November, the quantity of water at Mill Creek decreased from the maximum to 95 gallons per minute, most of which comes from strata near the surface and not from the tunnel itself, as there is practically no seepage in the tunnel at this point. Except for the few instances mentioned, where water under pressure was encountered, the working face is dry and seepage does

not begin for some time after the rounds are fired. At the west end the water drains by gravity, but as the Tye river incline was situated at a point where the cover was 97 feet, the water in the incline heading, before the pioneer tunnel was holed through, was drained by gravity to a sump at the foot of the incline and discharged by electrically operated Cameron pumps.

Air and electric power are used.

All of the mechanical operations in the tunnel are carried on by electric power, except the drilling and the operation of the shovels engaged in enlarging to full tunnel section which are done by air. The permanent trolley for the electric locomotives ends about 800 feet from the working face to prevent its being torn down by the force of the blasts from the shots. As the locomotive must work near the face, it is equipped with a special hook to fit over the trolley near the end, and the power is transmitted through a heavily insulated and very flexible wire which winds off a reel on the engine. As the wire winds off it puts tension in a spring and when the locomotive returns the spring automatically winds the wire on the reel. The power for the mucker and hoist is brought up also by extension wires to avoid damage to the permanent lines from the blast.

The drilling is done by means of air machines and water core drills. The drilling machines are mounted on the cross bars of a Denver Rock Drill Company's movable carriage which runs between the haulage tracks. As soon as the mucking operation is completed, the mucking machine is withdrawn, the track is laid to the face of the heading, and the drill carriage is shoved into place. Four or six drills are used simultaneously, depending on the size of the working face. Where the rock breaks easily, 10- and 12-foot rounds are pulled, but where the rock is seamy or inclined to break large, shallower holes or a larger number are drilled. The time required to drill a round varies with the character of the rock, and in some instances has required as much as six hours.

The track centers of the haulage tracks are such that the distance between the inside rails is equal to the gage of the tracks, so that the mucking machine and drill carriage run on what is to all intents and purposes a middle track and, therefore, are centered where brought up to the face of the heading.

By reference to the illustration it will be noted that the rear end of the drill carriage is equipped with manifolds to distribute the water and air to the several drilling machines so that only single lines of air and water need be connected when the carriage is placed for drilling. When it is spotted, it is anchored in place by means of a spud which depends from the forward end almost directly under the cross bar frame.

The mucking operations are carried on mechanically by means of Meyers-Whaley muckers in the headings, and steam shovels at the enlargement. Where the rock breaks in good shape, a 10 to 12 feet round can be mucked out in about 2½ h., while where the rock breaks large more time is required. Cars of 50 cubic feet capacity are used to transport the muck, and 12 of these cars are hauled at one time by General Electric tunnel locomotives. Only one car can be placed back of the mucking machine at a time and to avoid the delay of making a switch every time a car is loaded, a double track is laid to the face of the heading and the empties are set on one track, while the locomotive which is to dispose of the loads works on the other. A transfer hoist at the rear of the empty train lifts the cars over to the loading track and the locomotive merely shoves them back to the discharge of the mucker. This system reduces the time of mucking by about one-third and muck-

Fig. 5. — Timbered section of pioneer tunnel.

Flg. 6. — The drill carriage at the pioneer tunnel heading.

ing is often completed by the time the drillers have finished scaling.

At the west end where the enlargement has progressed beyond the first cross cut the material is being loaded on six-yard cars by a model 40 Marion shovel mounted on a caterpillar tractor, and the cars are handled to the dumping trestle by 20-ton General Electric locomotives. At Mill Creek a model 20 Marion shovel is being used in the full tunnel section west of the shaft. Electric locomotives and six-yard cars are in service at this face also.

A typical cycle of the heading operation at the Tye river incline, when the heading had been drifted 3 054 feet from the foot of the incline, is interesting :

Blast	7.10 a. m.
Time required for ventilation	27 minutes.
Time to enter drift, and clean up fly rock with mucking machine, and hand mucking	38 minutes.
Main muck pile.	1 h. 22 m.
Remove mucker and set drill carriage.	38 minutes.
Drilling 28 holes 8 ft. 6 in.	1 h. 17 m.
Loading and retiring	18 minutes.
Blast	11.50 a. m.
Total time of one complete cycle	4 h. 40 m.

Used 195 lb. 60 % gelatin.

Advanced 7 ft. 6 in.

Cross section 9 by 10 feet.

Four drills on Denver Rock drill carriage.

Character of rock : altered granite.

Number of men, 16 : including shifter and foreman of underground labor gang.

Twenty-eight 50-cubic foot Koppel cars loaded and dumped on waste pile at head of incline.

Bonus paid for rapid progress.

The high rate of progress has been mentioned. So far as known, the greatest monthly advance heretofore recorded was in the west pioneer of the Connaught tunnel at Rogers Pass on the Canadian Pacific in January 1915. While this advance was 932 feet, it was made on a face 8 feet by 6½ feet, or 52 square feet, whereas the high record of 1 157 feet at the Cascade tunnel was made on a heading 8 feet by 9 feet, having an area of 72 square feet, so that the amount of rock broken and removed was 38.5 % greater per running foot.

The heading operation is paid for on a footage basis with a bonus for progress in excess of a stipulated number of feet. Crews working directly in the heading are paid at different rates per foot of progress for the several services which they perform, but with the understanding that their pay will not be less than it would have been under the hourly rate which was in effect prior to 1 July. The footage price was established by using 650 feet as the basic progress for a month, and the established monthly pay on the hourly rate for a full month's work as the standard minimum. On this basis the bonus is derived by multiplying the number of feet advanced in excess of 650 feet by the footage rate. The second month this ruling was in effect the crews in the Scenic pioneer tunnel broke the world's record for advancing and greater progress was recorded in both September and October. All men directly engaged in the heading operation or working at the face come under the application of the bonus system, but the bonus rate is different for the different services and, as has been explained, depends on the hourly rates of pay in effect prior to 1 July. The several grades of employees that participate in the bonus are drillers and helpers, nippers, underground laborers and foremen, handy men hoist operators, mucker or shovel operators, brakemen, motormen and shifters, an average of 66 men in three crews at every face.

In addition to the bonus, a « high camp » pennant was designed and is given to the camp having the best record for the previous month. There is a friendly but none the less intense rivalry for the pennant, which has changed hands several times. The device on the pennant, the Great Northern « goat », is pertinent and is the cause of good natured gibes between camp adherents.

The following incident is indicative of the rivalry for the pennant, existing between camps. The footage basis of payment went into effect at 7.45 a.m., 1 July. The shift going off duty at this hour in the Tye river incline had a round drilled and loaded at 7.40. The progress had been so nearly equal that they had the choice of shooting before 7.45 and taking the pennant away from Berne, or holding the shot until after 7.45 and having the footage apply on the July bonus. They did not hesitate, but pulled the round and sacrificed the bonus for the pennant.

All gangs work eight hours, and those not engaged in the mining operations are paid either an hourly or a monthly rate. Men are not worked overtime unless absolutely necessary, but when they are required to do so, penalty overtime is not paid. All operations are carried on continuously three shifts a day, and seven days a week.

Source of electric power.

In a tunneling operation of such magnitude only air or electric power can be used to operate the mining machines and haulage equipment. When the work was started, no current was available at either portal although arrangements were made to bring power lines to these points as soon as weather and snow conditions permitted. At the west portal it was not possible, however, to complete the power line earlier than about the middle of the summer of 1926, and to avoid delay a generating station was established at Scenic. Two six-cylinder 360-H. P. and one two-cylinder 120-H. P. Fairbanks, Morse & Co., Diesel engines were installed to drive the generating equipment; delivery of power from this plant began 18 March and continued to 6 August, when the Great Northern power line from Everett, 97 miles, was completed. A maximum of 14 300 kw.-h. a day is used at the west portal in the mining and hauling operation, the compressor room, and the repair shop.

Electric traction is employed in the existing tunnel and the Great Northern constructed a power line from Cascade, the east portal, to a frequency changing station at Lanham Creek spur, from which the power was distributed to Berne and Mill Creek. This line and station were completed 17 April and power is still being delivered at a maximum rate of about 10 000 kw.-h. a day.

All drilling machines are air operated and complete compressor plants with electrically driven Sullivan angle compound compressors have been established at all three camps. The plant at Scenic contains five compressors of 950 cubic feet capacity each, which compress the air to 110 lb., but at the other camps the number of working faces is not so great and a smaller compressor capacity is employed, although space is provided for additional equipment when needed. The air for ventilation is supplied by electrically driven fans, and is conducted to the point of delivery through sheet metal pipes of the required size.

Storehouses and completely equipped repair shops, with machine tools, forges, and other equipment, are established at each camp, and a force of skilled mechanics is employed so that repairs can be made to practically every piece of equipment in use. A section of the shop is set aside especially for grinding and tempering drills.

Much care given to camp construction.

The camp buildings are substantially constructed and every effort has been made to give the men clean, sanitary and comfortable quarters. Bunk houses accommodate 10 men, and well equipped bath houses with showers and laundry facilities are placed at intervals in the camps to avoid crowding. Sanitary toilets and sewage disposal have been prov-

ided. Every camp has a commissary, a lounging room, and a recreation building where lectures, moving pictures or other entertainment may be given and dances held. Separate houses have been constructed for the men who have families and a monthly allowance is given those who board themselves. The dining rooms are spacious; the kitchens are exceptionally well equipped and have bakeries of ample capacity to bake all the bread and pastry used. Special houses have been built for the engineering staff and the contractor's supervisory forces.

The new alinement required the purchase of a summer resort hotel at Scenic and this has been partly remodeled to provide offices for the contractor and the engineering forces. The accommodations are sufficient to house about 50 employees and such visitors as have business with the railroad or contractor.

All three camps are in the mountains far from school accommodations and to provide instruction for the children in the camp, schoolhouses with sanitary toilets and heating plants have been erected to care for 50 children at Scenic and 35 each at Berne and Mill Creek. The local school authorities provide the teaching force.

Line changes.

The east portal is below the elevation of the present track at Berne so that a minor line change and grade reduction was required to reach the tunnel. The present alinement in this section is very crooked and, as a large amount of material which otherwise would have to be wasted is available for embankments, a revision of the line from Berne to Merritt, 9½ miles, is being made to reduce the heavy curvature and eliminate a number of snow sheds. The only improvement in the grade, however, is that brought about by the slightly lower elevation of the portal.

At Scenic, it was necessary to construct about two miles of new line to connect the tunnel with the present track. Descending to the west, the gradient is 1.7 % compensated for the first mile west of the tunnel to provide for yard tracks. The new line crosses the existing line twice, once immediately east of Scenic station, where it is 32 feet higher than the old line, and again near the junction of the two lines one mile farther west where the difference in elevation is 5 feet. To facilitate dumping the material from the tunnel, which is used to construct the embankment, a temporary pile trestle has been driven from the tunnel to the junction of the two lines. Directional passing sidings for east and westbound trains will be constructed at both ends.

Chronology and progress.

The work of clearing the sites for the camps began on 1 December 1925. The approach cut for the west pioneer tunnel was commenced on 14 December and on 29 December the adit or side drift, to reach the center heading at Berne, was started. The drift for the Tye River incline was begun on 23 January 1926, and the sinking of the Mill Creek shaft was started an 30 January. On 27 November 1926, the advance faces in the drifts had made the following penetrations :

West pioneer (including Tye River incline), Scenic	8 140 feet.
Center heading, Scenic (eastward) . .	3 498 —
Center heading, Berne (westward) . .	6 688 —
Center heading eastward from Mill Creek shaft.	1 512 —
Pioneer west from Mill Creek shaft. .	597 —
Full tunnel, Scenic (eastward) . . .	1 742 —
Full tunnel west from Mill Creek . .	447 —

The contract stipulates that the entire work shall be completed not later than 30 October 1928, and on 1 March 1926, after the preliminary work was finished, a progress chart delineating the progress which must be made to accomplish this

was prepared jointly by the railroad and contractor. On 27 November, the several operations were ahead or behind the schedule as follows :

—	Feet ahead.	Weeks.	Feet behind.
West pioneer (including Tye River incline), Scenic . . .	860	4 1/2	...
Center heading, Scenic	328	2	..
Centre heading, Berne	2 200	19	...
Center heading, east from Mill Creek shaft	425	3 1/2	...
Pioneer, west from Mill Creek shaft.	37	2/7	...
Full tunnel, Scenic	75	1/2	...
Full tunnel, west from Mill Creek shaft	...	1 1/2	90

Supervision.

A. Guthrie & Co., Inc., of St. Paul, have the contract for the tunnel work and line changes; J. C. Baxter, vice-president, has general charge for the contractor; while R. F. Hoffmark, general superintendent, has direct charge of the entire operation. W. E. Conroy is assistant general superintendent and H. J. King, F. J. Kane and C. G. Jones are camp superintendents at Scenic, Berne and Mill Creek, respectively.

The project is being carried on under the general direction of J. R. W. Davis, chief engineer of the Great Northern. Colonel Frederick Mears, assistant chief engineer, has direct charge for the railroad, and M. J. C. Andrews is resident engineer.

[621 .13 & 621 .335]

Electric locomotives :

A method of classifying, analysing and comparing their characteristics, (1)

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Figs. 1 and 2, p. 321 and 322.

(*The Engineer.*)

Introduction. — A vast amount of technical literature has been written about electric locomotives, their mechanical and electrical features, their performances in service, and their merits and demerits as compared with steam locomotives. This paper deals with an aspect of the subject which does not appear to have received attention, and that is a method by which electric locomo-

(1) Paper read before the Institution of Mechanical Engineers, London, 19 November 1926.

tives as well as steam locomotives can be classified into types on a common basis, so that their characteristics can be analysed and compared with each other and their respective merits deduced therefrom.

There is much about electric locomotives that is puzzling to the steam locomotive engineer. The are not external evidences of the latent power of the machines, and no external signs to determine the sphere of utility — that is, their suitability for passenger, goods or shunting services. With steam locomotives, the size of the boiler, its working pressure, the number and diameter of the cylinders, the wheel arrangement, the number and diameter of the coupled wheels, the presence or absence of a tender, all of these external marks establish the identity of the steam locomotive and enable one to gauge its power and sphere of utility. On electric locomotives these external marks are either absent or without significance. For example, *wheel arrangements* are useless for identifying or classifying electric locomotives, since very often the same wheel arrangement may be common to a number of types. Thus the 0-4-4-0 wheel arrangement is common to passenger, freight and shunting electric locomotives and to electric motor coaches. Some other method of classification must be resorted to, and the author proposes for this purpose the method described later, which is believed to be original. He hopes to show how it is possible to deduce and compare the relative merits of various types of electric locomotives, merits so radical in character and numerous in their variety that steam railway engineers find it difficult to concede their validity.

Leading characteristics of steam and electric locomotives. — In order to classify the characteristics, it is necessary to choose those that are common to both types, and thence deduce a basis of comparison. The characteristics of steam locomotives include tractive efforts, heating surfaces, grate areas, cylinder volumes, adhesive weights, total weights in working order, weight of motive power equipment and horse-power.

The characteristics of electric locomotives include tractive efforts, adhesive total weights in working order, weight of electric motive power equipment and horse-power.

For the purpose of this investigation, the following characteristics have been chosen common to both systems as being the most suitable :

HP = The average horse-power which, for this paper, has been taken as equal to twenty times the grate area in square feet for the steam locomotives and equal to the one-hour horse-power rating for the electric locomotives.

A = The adhesive weight in tons of the locomotives when in working order.

E = The weight in tons of the motive power equipment which, for the steam locomotive, includes the weight of the boiler, water, coal, steam pipes, cylinders, pistons, piston-rods and motion, and for the electric locomotive includes the weights of the motors and the electrical equipment.

W = The weight in tons of the locomotives in working order.

The absolute values of the above characteristics are useless for comparing steam and electric locomotives, but their relative values or *ratios* between them give the required basis of comparison. For instance, the ratio $\frac{HP}{A}$ bears typical values for passenger engines, other typical values for goods engines and still other typical values for shunting en-

gines. Further, these typical values are different in the case of steam locomotives from what they are in the case of electric locomotives, as will be shown later.

Classification of characteristics. — In order to carry out his investigations, the author collected data regarding some ninety-two steam and electric locomotives, classified them and calculated the values of the characteristics HP, A, E, and W, as well as the values of the various ratios subsisting between them, which are $\frac{HP}{A}$, $\frac{HP}{E}$, $\frac{HP}{W}$, $\frac{E}{W}$, $\frac{E}{A}$ and $\frac{A}{W}$. An abstract of the results is tabulated in table II, and a still further condensed abstract in table I.

TABLE I.

Average values of ratios $\frac{HP}{A}$, $\frac{HP}{W}$ and $\frac{A}{W}$ for steam and electric locomotives.

Type of locomotive.	Steam locomotives.			Electric locomotives (average of all systems).		
	$\frac{HP}{A}$	$\frac{HP}{W}$	$\frac{A}{W}$	$\frac{HP}{A}$	$\frac{HP}{W}$	$\frac{A}{W}$
Passenger	15.40	5.67	0.375	30.89	22.99	0.753
Goods.	10.96	5.70	0.528	19.18	18.24	0.940
Suburban motor coach	12.37	4.18	0.337	13.59	11.15	0.790
Shunting.	9.03	6.31	0.701	11.15	10.32	1

Only three of the ratios have been shown in the above table in order to make the issues more clear. The complete abstract of average values for all the six ratios referred to is given in table II.

It will be noted in table I, that the three ratios chosen are higher in every case for the electric locomotives than for the steam locomotives of the same type.

Also the ratio $\frac{HP}{A}$ is highest for the higher-speed passenger locomotives — both steam and electric — and lowest for the lower-speed shunting locomotives — both steam and electric. In other words, the ratio $\frac{HP}{A}$ is a measure of the speed for which the locomotive is designed, a fact which can be reasoned out as follows :

The horse-power of a locomotive is proportional to the product of its tractive effort and its speed, and the former is proportional to the adhesive weight of the locomotive. Consequently, the horse-power of the locomotive may be said to be proportional to the product of its adhesive weight and speed. Therefore, $\frac{HP}{A}$ is proportional to $\frac{A \times \text{speed}}{A}$, that is, proportional to the speed.

In the same way, it will be shown that the other five ratios, $\frac{HP}{E}$, $\frac{HP}{W}$, $\frac{E}{W}$, $\frac{E}{A}$, and $\frac{A}{W}$, bear typical values for steam passenger, goods and shunting locomotives, and

TABLE II.

Classification of characteristics of steam and electric locomotives (being the average value of ninety-two steam and electric locomotives).

Type of locomotive.	Average values of ratios.					
	$\frac{HP}{A}$	$\frac{HP}{E}$	$\frac{HP}{W}$	$\frac{E}{W}$	$\frac{E}{A}$	$\frac{A}{W}$
Passenger :						
Steam	15.4	12.85	5.67	0.443	1.201	0.375
S. P. S.	23.7	32.9	16.26	0.499	0.703	0.688
D. C. S.	24.78	46.5	18.8	0.404	0.521	0.78
T. P. S.	44.2	..	33.9	...	...	0.79
Goods :						
Steam	10.96	12.00	5.7	0.476	0.914	0.528
S. P. S.	14.34	27.06	12.03	0.459	0.546	0.85
D. C. S.	15.4	41.9	14.9	0.358	0.374	0.97
T. P. S.	27.8	...	27.8	...	...	1.00
Shunting :						
Steam	9.03	14.43	6.31	0.438	0.618	0.701
S. P. S.	10.53	21.90	10.53	0.482	0.482	1.00
D. C. S.	8.77	22.18	8.77	0.403	0.403	1.00
T. P. S.	14.15	30.9	14.15	0.475	0.475	1.00
Motor coach :						
Steam	12.37	17.4	4.18	0.24	0.711	0.337
S. P. S.	12.66	27.4	8.45	0.316	0.479	0.720
D. C. S.	14.53	51.4	12.2	0.218	0.290	0.85

NOTE.— 1. This table forms a key for identifying the type of any steam or electric locomotive by comparison of its characteristic ratios with those in the table.

2. It is quite possible that two new types of electric locomotives may be evolved in the near future, *i. e.*, the high-speed express passenger and the fast goods electric locomotive. These would have characteristic ratios differing from those shown in table II.

other typical values for electric passenger, goods and shunting locomotives. To show this, table II has been prepared. From the study of this table, the following conclusions may be drawn :

1. That the horse-power per ton of adhesive weight is much higher for electric than for steam locomotives of the same type.

2. That the horse-power per ton of electric equipment for the electric locomotives is much higher than the horse-power per ton of steam equipment for the steam locomotives.

3. That the horse-power per ton of total weight in working order is much higher for electric than for steam locomotives of the same type.

4. That the weight of the electric equipment is approximately the same as the weight of the steam equipment for locomotives of the same total weight.

5. That for locomotives of the same adhesive weight, the weight of the electric equipment is much less than that of the steam equipment.

6. That passenger electric locomotives have the greatest value of the ratio $\frac{HP}{A}$, $\frac{HP}{W}$ and $\frac{E}{A}$, and the least value of the ratio $\frac{A}{W}$, as compared with electric goods and shunting locomotives, whether direct-current, single-phase or three-phase.

7. That shunting electric locomotives have the least value of the ratios $\frac{HP}{A}$, $\frac{HP}{W}$ and $\frac{HP}{E}$, and the greatest value of the ratio $\frac{A}{W}$ as compared with electric passenger and goods locomotives, whether direct-current, single-phase or three-phase.

8. Electric motor coaches have the least values of the ratios $\frac{E}{W}$ and $\frac{E}{A}$ of all the types considered, whether direct-current, single-phase or three-phase.

9. Electric goods locomotives have ratios intermediate in value between those of passenger and shunting locomotives.

10. That of the electric locomotive types, the ratios for the single-phase system most resemble that of the steam, and the ratios for the three-phase system least resemble those of steam locomotives. Thus the determination and classification of the above ratios for any steam or electric locomotives furnishes a method of identifying its type or sphere of utility.

Analysis of characteristics. — Having evolved a common basis of classifying electric and steam locomotives into various types, to determine their sphere of utility, the author will now endeavour to analyse the characteristic ratios and to deduce therefrom the respective merits of 1. steam locomotives, 2. single-phase, 3. direct-current, and 4. three-phase locomotives. In what follows, the initials S. P. S., D. C. S. and T. P. S. designate single-phase system, direct-current system, and three-phase system respectively, and the values of the ratios $\frac{HP}{A}$, $\frac{HP}{E}$, $\frac{HP}{W}$, $\frac{E}{W}$, $\frac{E}{A}$ and $\frac{A}{W}$ for the above will now be examined in detail.

The ratio $\frac{HP}{A}$. — This might be called the speed efficiency ratio, because it is a measure of the speed for which the locomotive is designed, and it is therefore an index of the sphere of its utility. Table III is an abstract of the average values of this ratio.

TABLE III.

Average values of the ratio $\frac{HP}{A}$.

Type of locomotive.	Steam.	Electric S. P. S.	Electric D. C. S.	Electric T. P. S.
Passenger	15.4	23.7	24.78	44.2
Goods	10.96	14.34	15.40	27.8
Shunting	9.03	10.53	8.77	14.15
Motor coach . . .	12.37	12.66	14.53	...

NOTE. — The above are average values of this ratio, but the maximum values may be much higher, especially for high-speed express passenger locomotives, for which the ratio $\frac{HP}{A}$ may be = 50 or more. Also the tendency on electric railways in future will be to raise speeds of goods trains to those of passenger trains to increase the capacity of the railway and this will increase the ratio $\frac{HP}{A}$ for these fast goods locomotives.

The conclusions that can be drawn from table III are:

1. That the T. P. S. has by far the highest value of the ratio $\frac{PH}{A}$ of all types and the steam locomotive by far the lowest.

2. That the ratio $\frac{HP}{A}$ increases with the speed for which the locomotive is designed, being highest for passenger and lowest for shunting locomotives.

The reasons for the high value of this ratio $\frac{HP}{A}$ for T. P. S. locomotives are as follows :

a) The T. P. S. motor is considerably lighter than an S. P. S. or D. C. S. motor of equal horse-power;

b) The auxiliary electrical equipment required for the T. P. S. motors is lighter than that required for the S. P. S. and D. C. S.;

c) On the other hand, for a given duty or service, the capacity or rated horse-power of the T. P. S. motor has to be greater than that of an S. P. S. or D. C. S. motor, due to the constant speed property of the T. P. S. motor and the necessity for providing for peaks loads. This will be explained when discussing figure 1.

The reasons for the low value of the ratio $\frac{HP}{A}$ for steam locomotives are as follows :

a) The fact that, for a given horse-power the weight of the motive power equipment of a steam locomotive is considerably more than that of an electric locomotive;

b) The fact that part of the adhesive weight of a steam locomotive consists of revolving balance weights, whose action is to reduce the coefficient of adhesion of the steam locomotive. Consequently, the tractive effort that can be utilised is less with the steam than with the electric locomotive of equal adhesive weight, and therefore the horse-power that can be usefully applied is less also;

c) The fact that the torque or turning moment of the driving wheels is not constant throughout each revolution of the wheels, but varies, whereas that of the electric locomotive is even. Consequently, the tractive effort and therefore the horse-power that can be applied to the steam locomotive is less than that of the electric locomotive of equal adhesive weight.

The ratio $\frac{HP}{E}$. — This may be called *the efficiency of the motive power* ratio, for it measures the output or work done per unit of time for every ton of the motive power equipment. Table IV is an abstract of the average values of this ratio.

TABLE IV.

Average values of the ratio $\frac{HP}{E}$.

Type of locomotive.	Steam.	Electric S. P. S.	Electric D. C. S.	Electric T. P. S.
Passenger	12.85	32.9	46.5	...
Goods	12.00	27.06	41.9	...
Shunting	14.43	21.90	22.18	30.9
Motor coach . . .	17.40	27.40	51.40	...

Unfortunately, the author has not been able to obtain the weights of electrical equipment of three-phase locomotives, except for one locomotive. But it is well known that for equal horse-power the weight of a three-phase motor with its auxiliary regulating equipment, is less than that of either an S. P. S. or D. C. S. motor and equipment.

From table IV it may be inferred :

a) That the three-phase locomotive

has by far the highest value of the ratio $\frac{HP}{E}$ and the steam locomotive by far the lowest of all types;

b) That the shunting locomotives—electric—have the lowest values of the ratio $\frac{HP}{E}$;

c) Notwithstanding the fact that T. P. S. motors have to be designed for a higher horse-power capacity for the same service, the weight of the electrical equipment of a three-phase locomotive is less than that of an S. P. S. or D. C. S. locomotive designed for the same service. The reason for this is that the three-phase motors take the current straight from the overhead contact wires, without the intervention of transformers used for S. P. S. motors, and with regulating apparatus weighing less than that necessary for D. C. S. motors;

d) That the ratio $\frac{HP}{E}$ for a passenger electric locomotive is greater than that for a goods locomotive and still greater than that for a shunting locomotive. This is due to the fact that high-speed motors are used for the passenger locomotives and low-speed motors for goods and shunting locomotives, and for the same horse-power a high-speed motor is lighter than a low-speed motor.

The ratio $\frac{HP}{W}$. — This might be called the *power-weight utilisation* ratio, for it is an index of the power developed for every ton weight of the locomotive. Table V gives the average values of this ratio.

From this table the following may be deduced :

a) That the ratio $\frac{HP}{W}$ is by far the highest for the T. P. S. electric locomotive and by far the lowest for the steam locomotive.

b) That for the electric locomotive this ratio $\frac{HP}{W}$ is highest for the passenger and lowest for the shunting locomotive, the ratio being thus an index of the service for which the locomotive is designed.

TABLE V.

Average values of the ratio $\frac{HP}{W}$.

Type of locomotive.	Steam.	Electric S. P. S.	Electric D. C. S.	Electric T. P. S.
Passenger	5.67	16.26	18.8	33.9
Goods	5.70	12.03	14.9	27.8
Shunting	6.31	10.53	8.77	14.15
Motor coach . . .	4.18	8.45	12.2	...

The reasons for the higher value of this ratio for the electric locomotives are as follow : First, for equal weights of equipment, the horse-power developed by the electric equipment is much higher than that developed by the steam equipment. Secondly, the deadweight such as frames, wheels, tender, coal bunkers, water tanks, required to carry the steam equipment is heavier than that required to carry the electric equipment. Consequently, the value of HP is higher and that of W is lower for the electric locomotive, so that the value $\frac{HP}{W}$ is much higher.

3. Amongst the electric locomotives, the T. P. S. has by far the highest value of the ratio $\frac{HP}{W}$ for the reason that for the same maximum horse-power the T. P. S. motor is lighter than the others, and, moreover, requires more simple and lighter auxiliary electrical apparatus, so that the deadweight required to carry the electric equipment for the T. P. S. locomotive is also lighter.

The ratio $\frac{E}{W}$. — This might be called *the index of the utilisation of total weight by the weight of the motive power.* The higher this ratio the more effective may be said to be the motive power of the locomotive, for any given horse-power. In comparing this ratio for steam and electric locomotives, it should be borne in mind that for a given horse-power the value of E is far lower for electric than for steam locomotives. Consequently, the ratio $\frac{E}{W}$ will not materially differ.

The average values of this ratio $\frac{E}{W}$ are given in table VI.

TABLE VI.

Average values of the ratio $\frac{E}{W}$.

Type of locomotive.	Steam.	Electric S.P.S.	Electric D.C.S.	Electric T.P.S.
Passenger	0.443	0.499	0.404	...
Goods	0.476	0.459	0.358	...
Shunting	0.438	0.482	0.403	0.475
Motor coach. . .	0.240	0.316	0.218	...

The following may be deduced from this table :

a) That the ratio $\frac{E}{W}$ is lowest for motor coaches.

The reason for this is that the weight W includes not only the deadweight required to carry the electrical equipment, but also the deadweight required to carry the passengers. The other types do not carry passengers, whereas the motor coach does;

b) The ratio $\frac{E}{W}$ does not materially differ from steam and electric locomotives built for the same service, for the reasons already stated.

The ratio $\frac{E}{A}$. — This might be called the ratio of *utilisation of motive power weight by the adhesive weight.* When comparing this ratio for steam and electric locomotives, it should be borne in mind that the horse-power developed by the electrical equipment E is much greater than that developed by steam equipment of equal weight. The average values of this ratio for the various locomotives are given in table VII.

TABLE VII.

Average values of the ratio $\frac{E}{A}$.

Type of locomotive.	Steam.	Electric S.P.S.	Electric D.C.S.	Electric T.P.S.
Passenger	1.201	0.703	0.521	...
Goods	0.914	0.546	0.374	...
Shunting	0.618	0.482	0.403	0.475
Motor coach. . .	0.711	0.479	0.290	...

The deductions from this table are :

a) That the ratio $\frac{E}{A}$ is highest for passenger engines and lowest for motor coaches, except steam;

b) That the ratio $\frac{E}{A}$ is highest for steam locomotives and lower for D. C. S. and T. P. S. than for the S. P. S.

The ratio $\frac{A}{W}$. — This might be called *the ratio of utilisation of total weight by the adhesive weight.* It is therefore an index of the suitability of a locomotive for shunting, banking or heavy grade services for which the ratio $\frac{A}{W}$ should be as high as possible.

The average values of this ratio are given in table VIII.

TABLE VIII.

Average values of the ratio $\frac{A}{W}$.

Type of locomotive.	Steam.	Electric S. P. S.	Electric D. C. S.	Electric T. P. S.
Passenger	0.375	0.688	0.78	0.79
Goods	0.538	0.850	0.97	1
Shunting	0.701	1	1	1
Motor coach . . .	0.337	0.72	0.85	...

The deductions to be drawn from this table are :

a) That the ratio $\frac{A}{W}$ is highest for shunting engines and lowest for passenger engines.

b) That the ratio is highest for T. P. S. locomotives and lowest for the steam locomotives.

An important deduction that may be drawn from the study of the various tables is that the S. P. S. locomotive more resembles the steam locomotive in its properties than do either the D. C. S. locomotive or the T. P. S. locomotive. The latter is radically different in its properties from the steam locomotive, as will be explained, as it has a constant speed characteristic, whereas the steam locomotive has a constant horse-power characteristic.

Comparison of the characteristics of steam and electrical locomotives. — In order to compare the characteristics of steam locomotives with those of S. P. S., D. C. S. and T. P. S. locomotives, it will be assumed that a number of hypothetical locomotives have to be designed, one of each type. with the same wheel arrangement 0-4-4-0 type, the same adhesive weight, 40 tons; and the same axle load of 10 tons. The starting tractive effort will be assumed to be the same for each and equal to

$$\frac{\text{the adhesive weight}}{4} = 22\,400 \text{ lb.},$$

and the duty to be performed will be in each case to haul a trailing train load of 115 tons up a 3 % gradient at a speed of 16 miles per hour.

The tractive effort at this speed for this duty will be = 11 720 lb.

The horse-power at this speed for this duty will be = 500.

The above purely arbitrary assumptions have been taken to be the same for all locomotives to be compared, but in actual practice they would not be the same for all the locomotives, owing to their widely different characteristics. These conditions correspond roughly to those obtaining on the Kalka-Simla Railway, except that the locomotives considered in this comparison are approximately 66 % more powerful than the existing locomotives on the Kalka-Simla Railway.

Figure 1 has been got out to compare the respective tractive effort-speed curves of the hypothetical steam, single-phase, direct-current and the three-phase locomotive referred to above.

Horse-power curves and train resistance curves have also been added to the diagram to bring out the respective properties and merits of each type of locomotive, and to furnish the data for plotting figure 2.

Figure 2 has been got out from the above diagram by multiplying the speed in miles per hour by the train load in tons hauled at that speed by each locomotive, and plotting the products.

Thus the diagram shows the hauling capacity at various speeds for each of the locomotives considered. The speed at which this hauling capacity is a maximum has been termed the *critical speed* on the diagram.

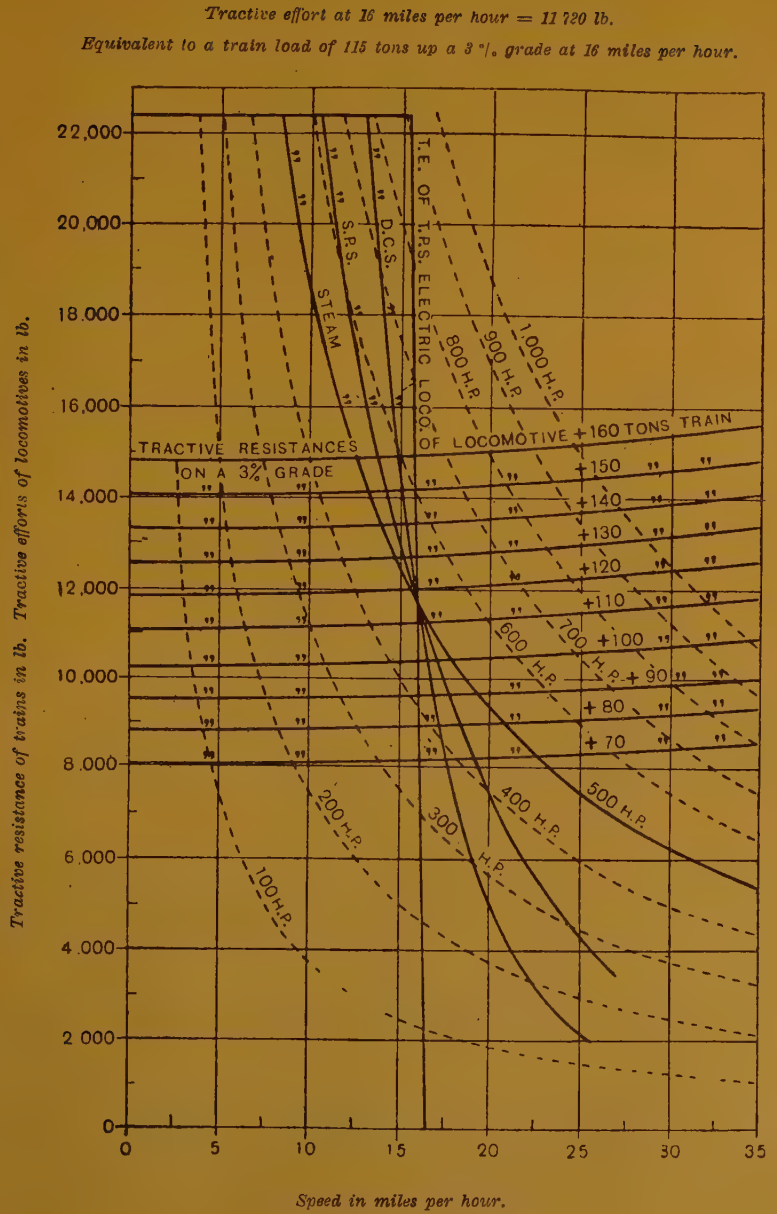

Fig. 1. — Comparison of tractive efforts of steam and electric locomotives of same adhesive weight, each being designed for the same service.

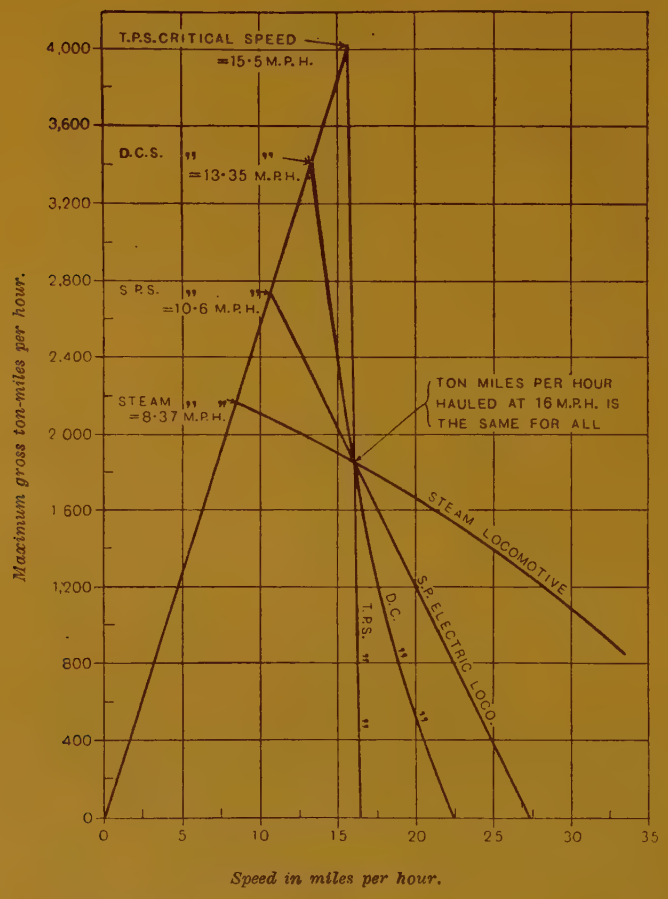

Fig. 2. — Maximum gross ton-miles per hour at various speeds which can be hauled by the steam and electric locomotives referred to on figure 1 up a 3 °/₀ grade.

The following conclusions are drawn from the analysis of the above diagrams and from the « ratios » :

A. — *Types of locomotives.*

1. Passenger locomotives have the highest values of the ratios $\frac{HP}{A}$, $\frac{HP}{E}$, $\frac{HP}{W}$ and $\frac{E}{A}$, and the lowest values of the ratio $\frac{A}{W}$.

2. Shunting locomotives have the highest values of the ratio $\frac{A}{W}$, and the lowest values of the ratios $\frac{HP}{A}$, $\frac{HP}{E}$ and $\frac{HP}{W}$.

3. Motor coaches have the lowest values of the ratio $\frac{E}{W}$.

4. Goods locomotives have intermediate values for all ratios.

B. — *Types of motive power.*

1. The three-phase electric locomotive has the greates maximum horse-power, the highest critical speed, greatest accelerating force start to full speed, and the least range of economical speeds and the highest values of the ratios $\frac{HP}{A}, \frac{HP}{E}, \frac{HP}{W}$ and $\frac{A}{W}$, and the lowest value of the ratio $\frac{E}{A}$.

2. The steam locomotive has the least maximum horse-power, lowest critical speed, least accelerating power start to full speed, and the greatest range of economical speeds. It has the highest value of the ratio $\frac{E}{A}$ and the lowest values of the ratios $\frac{HP}{A}, \frac{HP}{E}, \frac{HP}{W}$ and $\frac{A}{W}$.

3. The D. C. S. locomotive has properties and ratios intermediate between steam and three-phase locomotives, but closer to the latter.

4. The S. P. S. locomotive has properties and ratios intermediate between steam and three-phase locomotives, but closer to the former.

[621 116 & 669 .1]

The embrittlement of boiler plate, (1)

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Figs. 1 to 8, p. 326 to 328.

(*Railway Review.*)

Within recent years, a phenomenon of more or less frequent occurrence has become recognized, to which the term « the embrittlement of boiler plate » has been applied. Because of the relatively infrequent occurrence of this difficulty and the more or less obscure causes which bring it about, the fact of embrittlement has been largely in question, and both the evidence of its presence and the conditions which promote it are matters of importance at the present time. This is all the more accentuated by reason of the fact that seemingly within recent years an increased number of boiler failures chargeable to embrittlement are in evidence.

It should be said at the outset that these studies were undertaken by the engineering experiment station of the University of Illinois as a continuance of former studies, under conditions which would preclude any possibility of the work being done for any manufacturing

(1) Abstracted from a report presented at the convention of the American Society for Testing Materials, at Atlantic City, 21 to 25 June 1926.

or industrial enterprise, but solely for the purpose of developing information of general interest to all, whether concerned in the production of power or the fabrication of boilers.

It was determined at the outset that it seemed desirable to make a critical survey of the situation with a view to establishing beyond any question the facts as to the actual existence of such a thing as embrittlement. There has come somewhat persistently into the discussion of the topic references both to embrittlement as an actual fact and discussions making use of the expression « the embrittlement myth, » implying a question as to the reality of embrittlement. Hence the initial phase of these studies was directed towards proof of either the existence or the absence of the phenomenon.

In order to study the matter of embrittlement, therefore, it was deemed essential at the outset to develop some method of recognition which would be reliable without question, and distinctly differentiate such phenomena from other similar or related conditions. This led directly to a study of the cracking of rolled or fabricated plate. The studies along this line have revealed the fact that three types of cracks may occur as follows: corrosion cracks, fatigue cracks, and embrittlement cracks.

Corrosion cracks.

As the name implies, such cracks are due to direct corrosion of the metal. It is well known that where parts of the metal differ in composition or physical properties from other parts due to strains, density or impurities, there exists a difference of electric potential by reason of the fact that one portion of the metal is electro-positive to another. This electric potential sets up a galvanic circuit and in the presence of an electrolyte there is started a solution action on the positive side and a corrosion area is thereby developed. These corrosion areas or cracks in the case of metal under strain will follow the lines of stress as would naturally be expected. Also, since the electrolyte for producing the solvent action is accentuated or occurs in the presence of that type of ionization wherein the hydrogen ions predominate, it would be expected that such examples of corrosion would be found in the presence of such electrolytes as nitrates, chlorides, or sulfates, and such corrosion phenomena would be checked or inhibited by the presence of hydroxyl ions or that condition where an alkaline state exists. This in itself will explain why corrosion cracks would not be found in boiler waters which are alkaline in character. A study of chemical corrosion as described does not necessarily require a micrographic analysis of the structure of the corroded surface and the accompanying illustrations (fig. 1) include both the unmagnified and the magnified surfaces of cracks of this character. One specific feature should be noted in the micrograph, namely, the direction of the cracks follows the lines of stress, and this without regard to grain areas, that is, they proceed across the grains and disregard the grain boundaries in their path of development.

Fatigue cracks.

The development of cracks due to stress alone supplies interesting illustrative material in the study of metal cracks. Because of the abundance of material available of this type from the investigations of the fatigue of metals laboratory, an illustration is here introduced (fig. 2) because of its bearing on the general topic of cracking of plate. It seems that in metals placed under reverse strains, as failure is approached, cracks develop which upon micographic study reveal a positive characteristic which serves as a ready method of identification. By examination of a few type

specimens it is obvious that the cracks which have been started as a result of fatigue do not respect the fact of grain structure in the metal, but follow a course quite independent of grain boundaries and are hence described as cross-granular in character as opposed to inter-granular in their directional development.

Embrittlement cracks.

It is possible to differentiate those cracks which accompany embrittlement from either of the foregoing types for the reason that a micrographic study shows one distinctive characteristic which in all cases is found to accompany this type. By reference to the micrographs of embrittlement cracks, especialy when the surface is etched in such manner as to bring out clearly the grain boundaries, it is very evident that without exception these embrittlement cracks follow grain boundaries. Two typical micrographs illustrating the point are shown in figure 3.

It is clearly evident that certain regional areas may be defined in which embrittlement is more evident than in other regions where embrittlement does not exist. The waters of these regions are characterized by the fact of an almost complete absence of sulfates in the water, but this phenomenon accompanies another very marked characteristic and the one which is primarily responsible for the embrittling action, namely, the presence of free sodium bicarbonate.

The causes of embrittlement which can be traced to water treatment are comparatively few in number, but the possibilities of such distress if even remotely present should be thoroughly understood.

The embrittling effect is not confined to any particular make or type of boiler, all of the standard makes of both fire and water-tube boilers having been represented in the investigation.

Embrittlement is not confined to faulty or dirty iron, but quite as consistently occurs in iron of the best grade as otherwise. The micrographs in figure 6 will serve to illustrate these facts.

Summary of embrittlement characteristics.

The salient features which are characteristic of embrittlement may be more fully understood upon a brief survey of a few typical illustrations taken from the numerous cases which were noted.

Figure 7, for example, is a blow-off flange and it may be inferred, from the frequency with which cases of this sort are met, that the difficulty may be accentuated by the vibratory effect produced by the blowing off process.

Figure 8 is the outside of a drum from which twenty-six rivet heads were knocked off by sharp blows from a hammer. Figures 3 and 6 show micrographic studies of embrittlement cracks in these same or corresponding plates which have failed in service.

The characteristics of these cracks may be summarized as follows :

1. The cracks do not follow what is generally considered the line of maximum stress;
2. They start on what is termed the dry side of the plate;
3. They run, in general, from one rivet hole to another, though they often run past each other, leaving islands of plate;
4. They are irregular in direction;
5. They never extend into the body of the plate beyond the lap of the seam;
6. There is no elongation of the plate;
7. Where extreme action has occurred, rivet heads crack off or are easily dislodged by a slight blow of the hammer.

A summary of conditions relating to the location of the cracks in the boiler may be given as follows :

1. The cracks always occur below the practical water level;

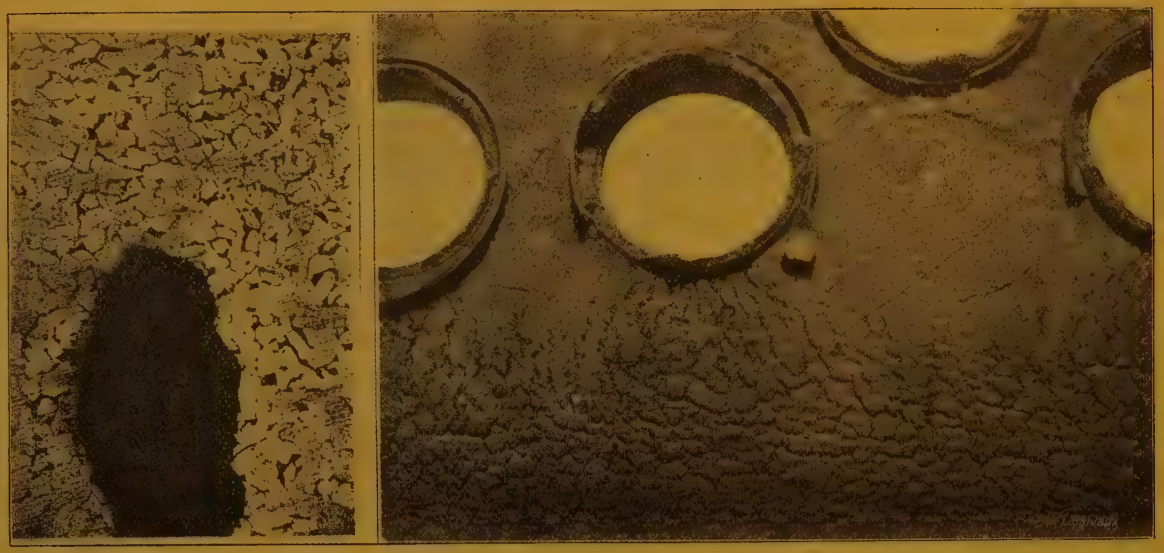


Fig. 1. — Corrosion cracking, etched section (× 42).

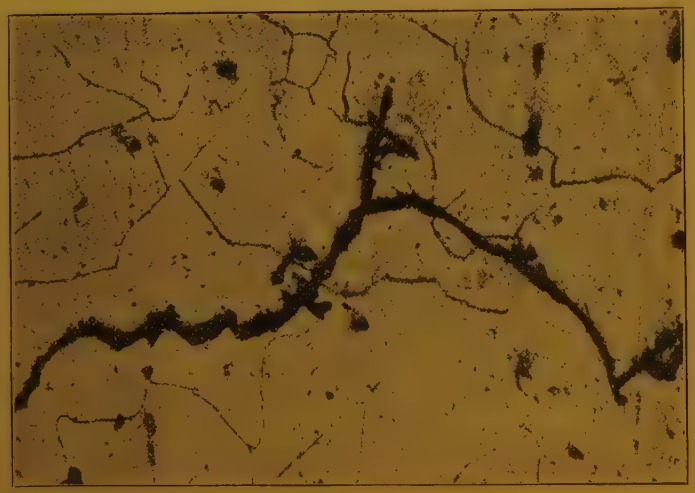
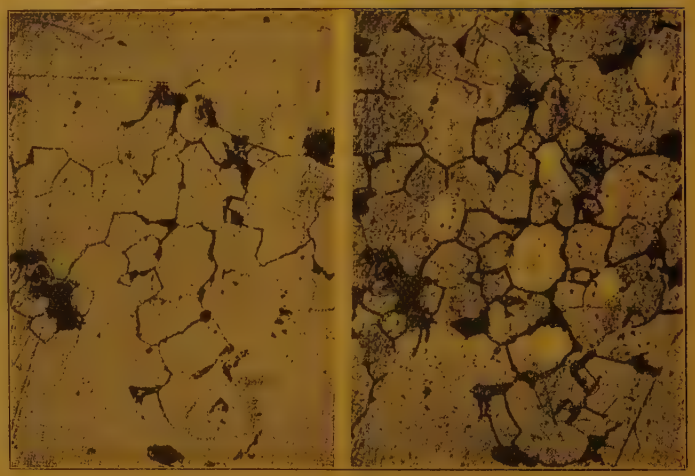


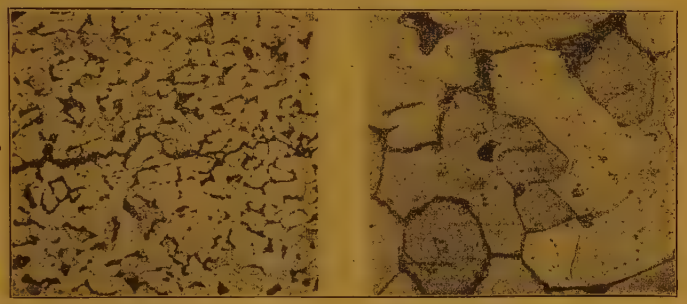
Fig. 2. — Micrograph of fatigue cracks in Armco iron (× 350), etched with 2 % nital solution.



(*a*) Unetched (× 250). (*b*) Etched (× 250)

Fig. 3. — Micrographs of embrittlement cracks from boilers.

Fig. 4. — Areas in which embrittlement occurs.



(*a*) (× 100) (*b*) (× 500)

Fig. 6. — Micrographs of cracks in embrittled plates.
Etched with 2 % nital solution.

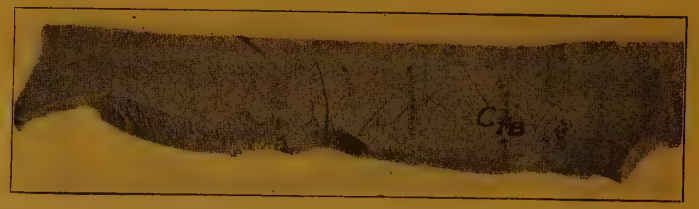


Fig. 5.
Section of cracked plate from boiler using treated water.

Fig. 7. — Embrittled blow-off flange.

2. All the cracks are in seams under tension;

3. They occur at places where the highest localized stresses might be assumed to occur;

4. The cracks occur in plates having

Fig. 8. — Outside of embrittled drum in a 500 H. P. boiler.

practically perfect chemical composition and physical properties as well as in plates of inferior make.

Table I gives the chemical analysis of boiler plate in which a few typical cases of embrittlement have occurred. It is of interest to compare these analyses with those of standard samples of high-grade material supplied direct from the manufacturers for testing under a later topic, that of « Laboratory reproduction of embrittlement ».

A summary of the chemical conditions which characterize the water used in boilers where embrittlement has occurred is as follows :

1. Sodium carbonate is the one substance which is always present in the feed water;

2. Sulfate hardness is usually absent or of a low ratio with respect to the sodium carbonate present. Sodium sulfate is similarly lower in amount than the sodium carbonate;

TABLE I.

Chemical analysis of plates from embrittled boilers.

—	Carbon.	Manganese.	Phosphorus.	Sulfur.
	Per cent.			
Bloomington, Ill. .	0.17	0.26	0.015	0.064
Hartland, Ill. . . .	0.22	0.39	0.017	0.024
Champaign, Ill. . .	0.20	0.50	0.005	0.024
Los Angeles, Calif.	0.22	0.54	0.013	0.036
Houston, Texas . .	0.26	0.34	0.014	0.025
Railway	0.14	0.37	0.018	0.018

3. Boilers encountering this trouble use waters having the characteristics noted under 1 and 2, and as a consequence of chemical reaction within the boiler develop a caustic condition with the sodium hydroxide in material excess over the sodium sulfate present.

Table II gives the analysis of well waters which have been used in embrittled boilers.

Laboratory reproduction of embrittlement.

It will be at once evident that if a method could be devised for reproducing the feature of embrittlement at will, a ready means would be available for determining both the cause of and the remedy for the difficulty. Intercrystallin cracking of mild steel is an abnormal type of failure; consequently if mild steel can be made to crack under conditions which may be controlled or modified at will and under conditions which parallel those of actual boiler operations, both the cause and the remedy may be in a fair way of solution.

TABLE II.

Analysis in grains per U. S. gallon of well waters which have been used in embrittled boilers.

—	Bloomington, Ill.	Urbana, and Champaign, Ill.	De Kalke, Ill.	McHenry County, Ill.	Watseka, Ill.	Paxton, Ill.	Sycamore, Ill.	Los Angeles, Calif.	Dallas, Texas.	Denver, Colo.	Houston, Texas.
Calcium carbonate . .	9.27	9.75	8.23	6.27	5.55	9.97	9.22	3 55	1.47	0.60	0.00
Magnesium carbonate.	6.82	6.34	4.72	5.18	2.99	6.64	6.26	0.38	1.01	0.26	7.1
Sodium carbonate . .	3.21	4.52	3.93	3.12	8.82	3.96	1.99	7.71	21.70	5.96	0.0
Sodium sulfate	0.00	0.10	0.08	0.00	0.83	2.36	0.16	5.72	11.92	1.49	2.58
Sodium chloride . . .	3.85	0.47	0.10	0.39	0.54	0.25	0.24	5.10	4.21	0.70	12.1
Iron oxyde and aluminum oxyde	0.12	0.10	0.07	0.53	0.09	0.03	0.17	...	...	0.05	0.0
Silica	0.58	0.82	0.40	0.23	0.31	1.82	1.18	0.95	2.51	0.63	...
Total solids. . .	26.35	23.17	17.92	16.03	20.04	27.80	19.45	...	48.50	9.69	...

The specimen to be tested was measured with a micrometer and the area calculated. The load necessary to give the required stress was calculated and the compression of the standardized spring to give this load determined. The specimen was set in the yoke and pinned to the plunger which was tightened by screwing down the nut on the upper end. When the spring was compressed to the right length the gland was tightened and the upper part of the apparatus placed on the container in which the desired solution had been previously placed. After the parts had been tightly bolted together they were placed in an electric furnace and heated until the desired pressure was obtained. A record of the pressure, temperature, and spring length was taken at regular intervals. When embrittlement had progressed sufficiently to break the specimen, the spring forced the plunger up, thus indicating that the specimen had broken. The release of the spring was in all cases found to be instantaneous and free from any indications of slow yielding.

TABLE III.

Chemical analysis of metals tested.

Reference.	Description.	Carbon.	Manganese.	Phosphorus.	Sulfur.	Silicon	Nickel.	Molybdenum.
		Per cent.						
F. S.	Flange Steel	0.180	0.450	0.012	0.027	...	...	...
Armco.	Armco Iron.	0.023	0.017	0.003	0.010	0.01	...	...
M. I.	Magnetic Iron	0.040	0.020	0.004	0.007	0.01	...	...
C. R.	Cold Rolled.	0.300	0.420	0.015	0.044	0.04	...	...
1112	S. A. E. 1112.	0.150	0.760	0.215	0.126	0.01	...	...
2312	S. A. E. 2312 (1) (2). . .	0.115	0.450	0.018	0.025	0.20	3.50	...
2212	Special (1) (2).	0.130	0 450	0.018	0.020	0.26	1.57	0.25
2330	S. A. E. 2330 (2). . . .	0.300	0.478	0.023	0.015	0.05	3.42	...

(1) Furnished by Central Steel Co., Massillon, Ohio.
(2) Cut to 1/4 inch flat from 1 1/4 inches round.

TABLE IV

Tension tests of materials used.

Reference.	Description.	Heat treatments (1).	Elastic limit, lb. per square inch.	Yield point, lb. per square inch.	Tensile strength, lb. per square inch.	Reduction of area, per cent.
F. S.	Flange Steel	As received	32 600	35 200	60 400	61
Armco.	Armco Iron.	Annealed 950° C. .	32 000	40 600	52 600	70
M. I.	Magnetic Iron . . .	— 950° C. .	25 000	30 000	50 700	72
C. R.	Cold Rolled.	As received	70 000	84 000	85 500	55
C. R. A.	Cold Rolled.	Annealed 850° C. .	...	36 400	64 700	55
1112	S. A. E. 1112.	— 900° C. .	47 500	49 000	74 000	56
2312	S. A. E. 2312. . . .	— 900° C. .	37 000	40 000	74 800	75
2212	Special	— 900° C. .	35 000	36 800	75 000	75
2330	S. A. E. 2330. . . .	— 850° C. .	41 000	49 200	77 300	57

(1) Annealed specimens heated to temperature indicated for 30 minutes and furnace cooled.

The average area of the test specimens was 0.05 square inch. The accuracy of the estimated load on the specimen was within 1 000 lb. per square inch when the stress was in the range of 10 000 lb. per square inch. In calculating the

stress, allowance was made for the stress added by the steam pressure acting on the plunger.

Test data on embrittling.

The results of the tests on the various metals embrittled are given in tables V to IX, inclusive. They indicate that two conditions must be present simultaneously to cause embrittlement of steel; first, the actual stress must be above the yield point of the metal; and second, the concentration of sodium hydroxide must be in excess of 350 gr. a litre (350 000 parts a million, 20 000 grains a gallon). The variation of pressure up to 200 lb. a square inch seems to have no marked effect on the rate of embrittlement.

When distilled water, sodium-carbonate solution, or sodium-sulfate solution was used in place of sodium hydroxide, the metal was not affected even when the stress was equal to that causing failure when sodium hydroxide was used.

The temperature of the previous complete annealing sample has no marked effect on the rate of embrittlement, neither has the amount of stress once it passes the yield point.

If the metal has been extremely cold-worked previous to testing for the rate of embrittlement, it will not lower the stress necessary to start this effect, but it appears that it will actually require a

TABLE V.

Effect on flange steel of stress and concentration of solution.

Solution, g. NaOH per litre.	Stress, lb. per square inch.	Container.	Time.		Gage pressure, lb. per square inch.
			Break.	No break.	
415	30 000	No. 1	...	22 days.	65
400	31 600	No. 4	...	18 —	65
400	33 000	No. 4	...	16 —	70
415	34 400	No. 1	...	27 —	90
400	35 400	No. 1	1 1/2 days.	...	60
400	39 200	No. 4	2 days.	...	55
400	44 300	No. 4	30 hours.	...	50
400	46 000	No. 2	1 1/2 days.	...	45
400	50 500	No. 4	27 hours.	...	50
40	50 500	No. 1	...	17 days.	55
200	47 000	No. 3	...	16 —	40
210	47 600	No. 3	...	21 —	65
345	46 000	No. 4	14 1/2 days.	...	65
405	50 000	No. 4	4 1/2 days.	...	50
400	39 200	No. 4	2 days.	...	55
455	52 000	No. 3	2 3/4 days.	...	95
575	46 700	No. 3	8 1/2 days.	...	50

TABLE VI.

Effect on flange steel of solutions of salts other than sodium hydroxyde.

Salt used.	Concentration, per litre.	Stress, lb. per square inch.	Container.	Time. No break.	Gage pressure, lb. per square inch.
None.	Distilled water.	48 600	No. 1	28 days.	50
Na_2COa	500	50 000	No. 3	34 —	70
Na_2SO_4	500	45 000	No. 4	30 —	70

TABLE VII.

Effect on flange steel of previous heat treatment, chemical action and pressure

Solution, g. NaOH per litre.	Stress, lb. per square inch.	Container.	Time.		Gage pressure, lb. per square inch.	Treatment.
			Break.	No break.		
400	39 200	No. 4	2 days.	...	55	No treatment.
400	50 000	No. 2	2 days.	...	55	Annealed 1050° C.
418	50 000	No. 2	20 hours.	...	55	No treatment.
407	50 000	No. 2	20 hours.	...	45	Dilute HCl 20 minutes.
407	50 000	No. 2	17 hours.	...	55	Lower half PtCl solution.
400	50 000	No. 2	36 hours.	...	55	All PtCl solution.
404	50 000	No. 4	...	16 days.	0 (1)	
400	46 000	No. 2	1 1/2 days.	...	45	
890	50 000	No. 2	2 days.	...	200	

(1) Open to air.

higher stress than would be required for the original unworked metal.

The change of chemical composition of the metal within the limits set for flange steel has very little effect on the rate of embrittlement. Thus, steels of the following general composition show no variation in behavior in respect to embrittling action.

Carbon	0.023 to 0.300 %
Manganese	0.017 to 0.450 %
Phosphorus	0.003 to 0.012 %
Sulfur	0.007 to 0.027 %

The introduction of 3.5 % nickel with the carbon either 0.115 or 0.3 % has no effect other than raising the yield point and consequently the initial stress necessary to start embrittlement. A combination of 1.5 % nickel with 0.25 % molybdenum behaves the same as nickel.

Table VIII.

Effect of cold work.

Specimen tested.	Stress before testing.	Stress during test, lb. per square inch.	Container	Time.		Gage pressure, lb. per square inch.
				Break.	No break.	
F. S.	...	39 200	No. 4	2 days.	...	50
F. S.	55 000 lb. per sq. inch.	25 000	No. 2	...	22 days (1)	50
F. S.	Cold worked 20 % elongation continued	25 000	No. 1	...	27 days.	55
F. S.	...	45 000	No. 1	12 hours.	...	60
C. R.	...	25 000	No. 2	...	24 days.	85
C. R.	...	38 000	No. 2	...	24 days.	85
C. R.	...	45 000	No. 2	17 1/2 days.	...	80
C. R.	...	55 000	No. 2	28 hours.	...	85
C. R. A.	..	38 000	No. 2	36 hours.	...	85

(1) Tested for tensile strength after 22 days in solution. Tensile strength 69 200 lb. per square inch.; reduction of area 52 %.

Table IX.

Effect of chemical composition.

Specimen tested.	Solution, g. NaOH per litre	Stress, lb. per square inch.	Container.	Time.		Gage pressure. lb. per square inch.	Yield point, lb. per square inch.
				Break.	No break.		
Armco.	409	40 200	No. 2	...	14 days.	75	40 600
Armco	400	45 000	No. 2	21 hours.	...	70	40 600
M. I.	405	35 000	No. 2	23 hours.	...	50	30 000
F. S.	418	50 000	No. 2	20 hours.	...	60	35 200
1112	409	55 000	No. 2	3 1/2 hours.	...	50	49 000
1112	400	40 000	No. 2	15 hours.	...	80	49 000
1112	427	35 000	No. 4	...	20 days.	80	49 000
1112	427	25 000	No. 4	...	25 days.	85	49 000
2312	375	48 000	No. 2	29 hours.	...	80	40 000
2312	450	43 800	No. 4	8 1/2 days.	...	85	40 000
2212	385	45 000	No. 2	20 hours.	...	65	36 800
2330	375	55 000	No. 4	3 1/2 days.	...	65	49 000

When the sulfur becomes 0.215 % and the phosphorus 0.126 %, the rate of embrittlement shows a marked acceleration.

The embrittled specimens are not corroded but are covered with a thin, shiny, blue-black, coherent coat of magnetic oxid of iron. They show no marked elongation or reduction of area at the break.

Micro-examination of embrittled metals.

Samples of cracked steel were obtained from different instances of this type of boiler distress. These when examined under the microscope showed that the cracks were intercrystallin. The manner in which mild steel fails under ordinary stress, either static or fatigue, is almost invariably trans-crystallin. The fact that embrittlement cracks progress between the grains indicates that this is not a normal fracture. Furthermore, the crack progresses without any marked deformation of the grain, another deviation from the general behavior of mild-steel static failures. When examined under higher magnification, the crack is observed to pass between the smaller grains of pearlite which make up the carbon areas.

The specimens embrittled in the laboratory have been examined and found to have a large number of cracks progressing into the strained portion and all of these are intercrystallin without any marked deformation of the grain.

Inhibition of embrittlement.

Since embrittlement is the result of the combined action of chemicals and stress on the metal, it appears possible to stop it by at least one or two methods. The stress may be reduced to a point where embrittlement will not occur. This might be possible if the actual concentrated stresses such as at rivet heads, around rivet holes, etc., could be calculated and the boiler constructed so as to keep these stresses low enough. Even this could not be an absolute prevention since some unknown localized stresses might still exist and meet the embrittlement conditions.

The removal of the cause of the chemical action, in the absence of which embrittlement would not occur, would be the best procedure; but even this is difficult in some cases by the fact that a change of water is not always possible. A close approach could be made by neutralizing the alkalinity with some chemical, but complete neutralization would be impossible without endangering the boiler.

When it is noticed that embrittlement never has occurred even in carbonate waters when the sulfate content is high, the question is raised as to whether or not the existence of a definite ratio between the sodium sulfate and the sodium hydroxide is sufficient in itself to prevent this cracking.

The effect of a sulfate-carbonate ratio in plant operation has been studied in one plant for a period of ten years. This plant, the University of Illinois power plant, had been experiencing embrittlement troubles for some time and when in 1915 three new drums had to be replaced after only five years of service the following system of water treatment was inaugurated : The sulfate-carbonate ratio of the feed water is maintained at two. This is done by neutralizing about seventy % of the alkalinity with sulfuric acid. The water is treated in two 40 000-gallon settling tanks. About 110 lb. of lime is added to each tank and after sufficient agitation and settling, the required amount of acid. Analyses of each tank are made daily after lime treatment, and again after the acid is added, to determine total alkalinity. Daily analyses are also made of the water in each boiler for causticity and

total alkalinity. After ten years of operation on this treatment the boilers were given a thorough inspection during February, 1926. Test rivets were removed and a close inspection made for signs of leaking or cracks around the rivet holes. At that time the boiler inspector from the Hartford Steam Boiler Inspection and Insurance Co. pronounced the drums to be in perfect condition. The rivets were redriven and the boilers are back in operation.

A power plant in Champaign had boilers go into service in 1916 using the same type of water as the University of Illinois but without the acid treatment, and in 1925 considerable trouble was experienced due to embrittlement. Figure 8 shows one of the condemned drums from this plant. These two plants operating for nine and ten years, within 20 lb. of the same steam pressure, and upon precisely the same type of water, one sulfate-treated and the other not, serve as a long-time experiment with results that are strictly in accord with the laboratory indications. The boilers using the treated water are in excellent condition while the others have been condemned after only nine years of service.

Cause of embrittlement in steam boilers.

It has already been shown that the production of intercrystallin cracks in mild steel can be brought about only by the combined action of stress and proper chemical attack. The limits set for each apply only to the conditions of the laboratory tests and the time of the tests. There is a possibility of a slower action taking place over a period longer than has been tested for, which may lower the stress or the concentration of solution necessary to some degree. Another point to be considered is that the embrittlement tests were run with the metal surrounded by a solution, while in boilers the solution is penetrating between the plates in a thin film and the limits of concentration of solution necessary to produce embrittlement may be altogether different. It is possible to conceive of the stress passing the yield point and the solution reaching the high concentrations used in the tests when one considers that embrittlement only occurs in the riveted areas which are under tension.

Probably in all boiler plates the stresses at the edge of rivet holes occasionally reach the yield point of the metal. It should be remembered that the mathematical theory of elasticity indicates that the localized stresses at the edge of a hole in a plate may be nearly three times the average stress in the plate. This stress concentration at the edge of holes, such as rivet holes, would in itself be sufficient to account for stresses almost high enough to start embrittlement should a solution of proper concentration penetrate to the rivet holes. In addition it should be noted that during the fabrication of the boiler the seams are put together by riveting. The riveting is done hot and, granting only sufficient riveting pressure is used to maintain tight seams, very high localized stress occurs under the rivet heads and around the rivet holes. When the appearance of the plates and straps is observed after they have been removed from a boiler which has been in service and the rings around the holes where the straps and plates have been held together by the rivets are seen, there remains little doubt that the metal has been stressed beyond the yield point. When this stress is combined with the stress incident to operation of the boiler, additional local stresses develop which may still more favor embrittlement. The sharp angle which the rivet head makes with the shank combined with the stress forcing the rivet heads apart undoubtedly produces sufficient stress at this point to allow embrittlement to start.

It is not uncommon to have rivet heads drop off embrittled boilers long before the plates are embrittled sufficiently to detect cracks. The stresses in plates caused by riveting were studied by R. Bauman, who showed that the plates are stressed locally to values exceeding the yield point even when low riveting pressures were used. H. F. Moore in his address before the American Society for Steel Treating in September, 1925, stated that there was no doubt of the existence of localized stresses which exceeded the yield point of the metal around the rivet holes in riveted sections.

This concentration of stresses exists in all boilers to a greater or smaller degree, yet in the absence of the proper chemical in the boiler water no embrittlement results. If the proper chemical, sodium hydroxide in the absence of much sulfates or carbonates, exists in the boiler water there is a possibility that sooner or later that boiler will become embrittled. The concentration of sodium hydroxide in the boiler seldom surpassed 17 000 parts per million or about 1 000 grains a gallon. Laboratory tests show that solutions of this concentration are not detrimental to the steel. There is a possibility that the actual concentration of caustic soda in the vicinity of the seam does sometimes reach a value much above this but still far below 400 000 parts per million or about 23 000 grains per gallon. It is known that sodium hydroxide even of such a low concentration as met with in boiler operation cannot readily be kept under pressure in containers with screw connections. Solutions of other salts can be retained, but with caustic soda leaks tend to develop. The presence of caustic soda under pressure in the boiler tends to produce a seepage of the solution into the seam between the metal surfaces. As the solution penetrates between the plates there is a tendency for concentration due to the steam being released back to the boiler or escaping to the outside. As the concentration progresses, the penetration of the solution further into the seam continues. With time, the concentration becomes sufficient to start embrittlement and if localized stresses are sufficient at the point of concentration embrittlement starts. Often this concentration is reached and embrittlement does not progress, apparently due to lack of sufficient strains and the penetration progresses until a free leak develops. On examination, no presence of embrittlement can be found. At other times embrittlement develops, the plates are weakened and a leak develops which indicates trouble. When the plates are removed from the seams of embrittled boilers soluble salts are found between the plates. This alone would indicate that there must have been considerable concentration in the seams. Another evidence of chemical attack in the seams is the presence of finely divided black magnetic oxide deposited on the surface of the plates.

The fact that the two main factors for embrittlement must occur simultaneously makes it possible for some boilers to operate under conditions which might point toward embrittlement without developing any distress. The fact that no distress (apparent by general inspection) has occurred is no sign that the boiler is free from danger.

The real cause of embrittlement in steam boilers can be summarized, therefore, as concentration of caustic soda in the seams in the absence of much sulfate or carbonate together with the existence of localized stresses in the seams.

The results of the experimental work on the prevention or inhibition of embrittlement indicate that it can be stopped, but at present these experiments are not complete enough to cite final ratios and concentrations of the chemicals involved. The data collected from plant

operation point out that the maintaining of certain ratios of the chemicals will prevent this trouble.

The removal of stresses is not to be considered in a commercial installation, and if it were possible it would not be applied to the many plants already in operation. The practice of inside caulking in drums will, if it can entirely prevent seepage into the seams, tend to retard the occurrence of this type of distress.

The removal of the source of the sodium hydroxide by changing the supply water would be the most effective of any method, but in some instances this is not possible and in others would lead to the use of waters which produce other boiler complications.

The removal of a large percentage of the sodium carbonate with the building up of a sodium-sulfate sodium-carbonate ratio either by the use of sulfuric acid, aluminum or magnesium sulfate all have their disadvantages. The acid treatment is not to be recommended unless installed under the advice and supervision of an expert in such matters and operated under supervision of a chemist or an engineer who understands all the possibilities of its danger. The aluminum and magnesium-sulfate treatment is very effective when used in connection with settling tanks and filters which remove the possibility of scale-forming ingredients in the boiler. These two chemicals are as dangerous, if used in excess, as is the use of acid and should be installed only after a thorough checking of the necessary amounts by a chemist who understands the reactions involved.

One point of particular interest is the fact that undecomposed sodium carbonate acts as an inhibitant. Thus if a plant should operate with a high percentage of make-up high in carbonates, the amount of hydroxide formed may be lower since the introduction of fresh carbonates would tend to keep the carbonate-hydroxide ratio high and may in turn inhibit embrittlement. Cases have been cited where the sodium-sulfate to sodium-carbonate ratio has been exceedingly low without any indications of embrittlement and the neutral effect of sodium carbonate is the obvious explanation.

The Bloomington plant is now operating on a sulfate water treated with lime and soda ash and illustrates the high sodium sulfate due to the reaction of calcium sulfate and sodium carbonate during the softening. The Chicago station and the University of Illinois are carbonate waters treated with sulfuric acid. The ratio of sodium sulfate to sodium carbonate is about 2 in both cases. The Chicago station working with a lower make-up and a higher steam pressure has a lower amount of remaining sodium carbonate than the University of Illinois water. Assuming that sodium carbonate is also an inhibitant the ratio from actual boiler analysis should be sodium sulfate plus sodium carbonate to sodium hydroxide. When this is the condition, the lower pressure plant with higher make-up is apparently more protected and could in turn work with a lower sodium-sulfate to sodium-carbonate ratio. The ratio of sodium sulfate to sodium carbonate will be misleading in a few isolated cases. The Illinois Central Railroad locomotives operating locally in Champaign on alkaline water illustrates this since the sulfate-carbonate ratio is shown to be as low as 0.63 but the sodium-sulfate plus sodium-carbonate to sodium-hydroxide ratio is over two which is much safer and apparently sufficient to stop embrittlement. These illustrations will serve to show that each plant must be considered by itself in order to set any prescription for feed-water treatment in order to inhibit embrittlement.

[656 .215 (.73)]

Floodlighting brightens classification yards, (1)

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Figs. 1 to 7, pp. 340 to 344.

(*Railway Review*.)

The lighting of railway classification yards is a problem of space lighting, the length of yards varying from 3 200 to 6 500 feet, the width from 400 to 650 feet, and the distance from the hump to the last switch on the ladder, from 1 300 to 1 800 feet in our modern yards, — all depending upon the capacity of the yard. Ten tracks generally diverge from each ladder track. The space between the ladder tracks is used for the speeder cars which carry the hump riders back to the hump after cars have been brought to rest in the classification yard. In the case of a yard where car retarders are used, this space could be utilized for pole lines carrying a lighting distribution system.

No work other than the classification of cars is performed in the modern classification yard, the air testing and inspection being performed in the departure yard, and consequently there is no necessity for local or intense illumination.

The ladder tracks of classification yards are seldom built on curves; in fact, in yards where retarders are used, the curvature is limited to six degrees. Therefore the problem of curved track lighting need not be considered.

There are two adjuncts to a modern classification yard; namely, the receiving yard and the departure yard. No illumination is ordinarily required in these other than that which is necessary for police protection or in cases where short trains are pushed into the receiving yard on the same tracks or diverging tracks, for the storage of previous trains. In some instances, however, where receiving yards are built with considerable curvature, there is an advantage in providing sufficient illumination to enable engine drivers to observe their respective courses.

The tracks connecting the classification and departure yards, covering a distance varying from 400 to 700 feet, require local illumination. The most effective means of providing this illumination is by the angular type of reflector, with the beam of light directed in such a manner as not to affect the vision of the engine driver who is required to cover these tracks and who is frequently unable to obtain the benefit of the locomotive headlight because of the curvature of tracks and on account of switches. With the angular reflectors giving a wide beam, the proper spacing is about 100 feet for 300-watt lamps at an elevation of 22 feet above the tracks. A similar type of lighting should be installed at the hump for the benefit of the car riders. The departure yard, as stated above, re-

(1) Abstract of a paper presented before the New York regional meeting of the American Institute of Electrical Engineers, 12 November 1926.

quires very little illumination other than that necessary for police protection. The inspectors are required to work under cars and between long lines of cars on adjacent tracks and therefore they carry lanterns. Moreover, it would be impractical and expensive to attempt to provide illumination for their benefit, and it would be impossible to show any financial saving or reduction of personal injury hazard.

General principles of lighting.

Good illumination requires an even distribution of light without glare. In order to obtain this condition, lamps must be so spaced and so placed that there is no interference with one's vision. As space is valuable in a railroad yard, and as the expense of installing and maintaining such an installation is extremely heavy, we have been compelled to seek some other solution of the problem. The method commonly used is the installation of towers 75 to 120 feet above the base of the rail, with a number of projectors equipped with 750-watt or 1 000-watt lamps mounted on a platform at the top of the tower. The protective apparatus and switches are installed at the base of the tower in order to avoid all unnecessary maintenance labor. Practically the only necessity for climbing the towers is to change lamps and clean lenses and reflectors, which it is rarely necessary to do more often than every three months in winter and less frequently in summer. It is advisable to place control of the lighting in charge of the hump tower operator. This may be accomplished by installing a small start and stop button in a selected position, the manipulation of which energizes and de-energizes a contactor installed at the lighting tower. Another method is to install time switches at the base of towers which will automatically turn the lamps on and off at a predetermined time. The first method of control enables the tower operator to practice economy in current consumption during slack hours and yet furnishes sufficient light for police purposes. Tower lighting permits adjusting the position of the towers with the result that if space is not available for an ideal location, a change of 100 feet or even 200 feet does not materially decrease the efficiency of the installation. There is a possibility (although rarely available in classification yards) of taking advantage of high structures such as coal pockets and bridges for mounting projectors.

Classification yard lighting may be divided into three general classes or types, each requiring a different solution, as follows :

1. Yards employing hump riders and having hand-operated switches;
2. Yards employing hump riders and equipped with automatic switches;
3. Yards equipped with automatic switches and car retarders.

Lighting for yards in which hand switches and hump riders are employed.

With the first type of yard, extreme care must be exercised to prevent interference with the switchman's vision on account of glare, as this is likely to be the cause of personal injury or accidents to the equipment. If space is available, two towers may be installed one on each side of the hump and 300 feet apart. If lack of space at the hump makes this impracticable, a so-called « modified distribution system » is advisable and the towers may be installed on the outer edges of the classification yard in conjunction with 300-watt angular reflectors mounted on poles in the space occupied by speeder tracks which provide illumination in the area between the hump and the point where the tower lighting becomes effective. In very narrow yards it may be found desirable to install 500-watt projectors with a 250-foot spac-

Fig. 1. — Illustration of visibility secured by silhouette and specular reflection from the rails and sides of cars.

Fig. 2. — Daylight view of the Cedar Hill yard of the New Haven.

Fig. 3. — The Cedar Hill yard of the New Haven as it looks when lighted at night.

ing and an elevation of from 35 to 40 feet above the tracks, the necessary pole line being installed in the speeder car space. This latter method does not permit expansion and involves high maintenance costs as well as interference with the operators' vision by the poles; therefore it should not be installed. The cost of installing two towers at the hump amounts to approximately $7 000, and involves a low maintenance cost, whereas the « modified distribution system » costs at least $8 000 to install and results in a high maintenance cost.

Yards having automatic switches and employing hump riders.

With the second type of yard it is not necessary to consider the glare as there are no men in the yard and the current of traffic is in the direction of the light beam. One or two towers may be installed in the same relative position as in the first type of yard, or, if space is not available at that point, they may be installed close to the hump. The cost of these towers will be approximately $7 000. For a 3 200-foot yard no other lighting is required for the classification yard, an unobstructed view being obtained both from the hump and by the car riders. The power required for a 40-track yard is approximately 16 kw. The installation and maintenance cost is low. The maximum illumination is at the switches, and the hump rider is enabled to detect track conditions clearly; moreover, in distant parts of the yard sufficient illumination is provided to permit the observation of low built cars in time to avoid collision with them.

Where yards are of extreme length it will be necessary to place at least two towers approximately 2 000 to 3 000 feet from the hump in the yard, or where space is not available for this, the towers may be installed on the outer edges of the yard at the same distance from the hump. The wattage requirements for a

Fig. 4. — Modified lighting system where space at sides of hump is limited.

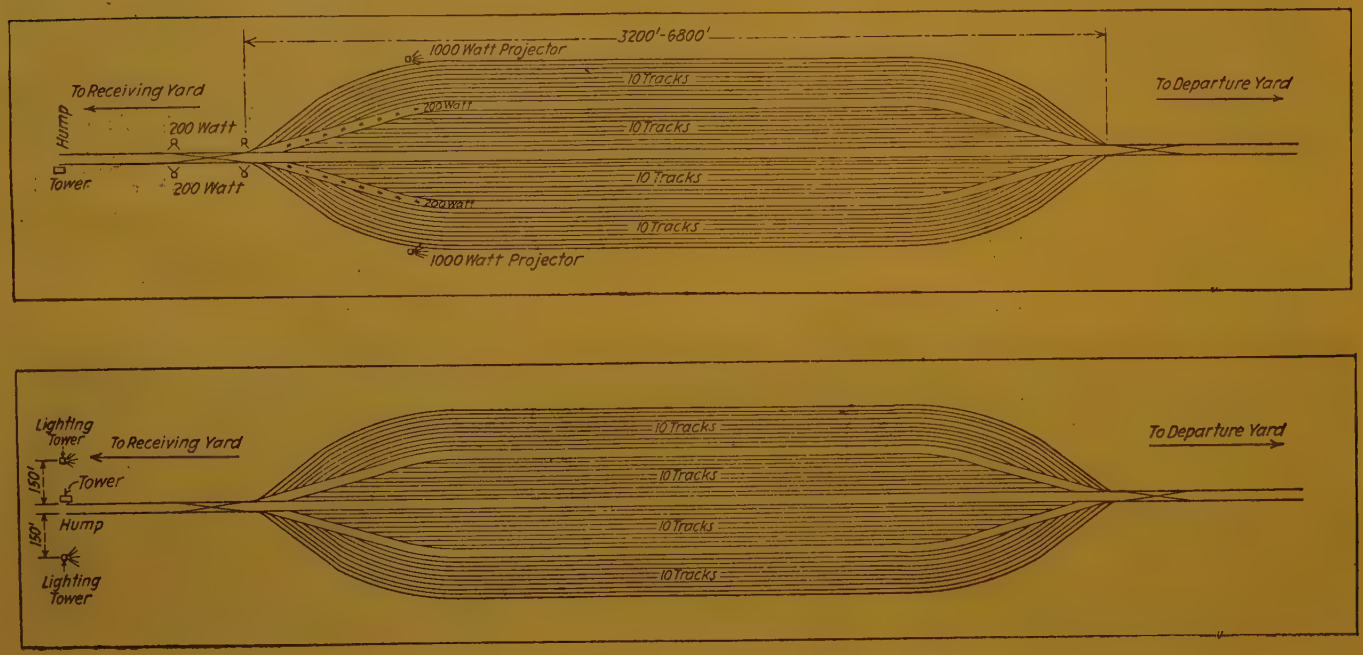

Fig. 5. — Typical lighting with towers located at the hump.

Fig. 6. — Two 70-foot towers, each equipped with eight 1 000-watt projectors and located close to the hump used for lighting the 3 200-foot Cedar Hill, Conn., yard of the New York New Haven and Hartford.

Fig. 7. — Yard equipped with car retarders showing relative position of operating towers and floodlight tower.

3 200-foot yard are from 12 to 16 kw. and decrease proportionately as the yard decreases in length. The increase in installation cost for a 6 000-foot yard over a 3 200-foot yard will amount to approximately $12 000, making a total installation cost of about $19 000, with a maintenance increase of slightly over 100 %. There is some possibility that at some time in the future, lamps of higher wattage may be used in the 5 000- and 6 000-foot yards, with a resultant reduction in the number of towers required and a decrease in the installation and maintenance costs, but in considering any such proposition the possibility of interference with the vision of adjacent interests must be taken into account.

Yards in which car retarders are used.

The third type of yard is similar to the first, but the personal injury hazard is eliminated as no hump riders or switchmen are required in the yard.

The retarder operators are located in secondary towers situated adjacent to the groups of retarders which they control. These towers are in the area of the ladder switches and are at an elevation above the tracks which permits an unobstructed view of the yard. Up to the present time no economical solution of this problem has been worked out. If lighting as recommended for the second type of yard is installed, the glare interferes with the vision of the operators. In this connection it is possible that a visor effect may be obtained by painting the upper part of the tower windows, thereby shutting off from the operators' vision the direct beam of the lights.

The position of retarders with respect to the operator is frequently such that it is difficult for the operator to observe the exact position of moving cars and of retarders at night on account of shadows. A possible solution of this problem may be found in « marker lights » so placed that they will indicate when the braking effect on cars is to be applied to the retarders. This solution of the problem would be very economical.

Any system of local illumination covering retarder areas would be expensive and would cause interference with track clearness, which must be avoided.

With the increased experience of retarder operators there will be no necessity for illumination greater than that provided in yards using hump riders and automatic switches, since each operator will know the loading condition of his tracks and will be governed accordingly. The same problem presents itself in lift-bridge operation, where actual practice has demonstrated that the experienced operator knows the position of his bridge at night and does not watch the indication lights, but prefers a marker light so placed that it will not be necessary for him to direct his attention in another direction.

Advantages of yard lighting.

The committee on illumination of the Association of Railway Electrical Engineers in November, 1923, reported the following advantages of yard lighting :

1. Speeding up of cars handled in the yard at night;
2. Reduction in cars damaged by rough handling and collision in the classification yard with consequent reduction in claims, delay in delivery of goods, loss of service of damaged cars, etc.;
3. Reduction in losses due to pilfering on account of more effective policing possible with a well illuminated yard;
4. Improved working conditions and increased safety for employees working in the yard.

The studies and investigations of the committee on illumination in conjunction with the lamp and projector engineers, although not as yet completed or published, have already resulted in more efficient units and more economical installations. The problem of lenses de-

teriorating with age, with resultant decrease of efficiency, has been practically solved. Curves and engineering data are now available which makes it possible to plan installations with confidence that the results anticipated will be achieved. The problems of space lighting may be solved in various ways on account of local conditions, but the fundamental principles are the same in all instances.

Floodlighting is not a « cure-all », but the results are satisfactory to the operating people and provide for them the operating conditions which they have been seeking.

No mention has been made in this discussion of silhouette effect, since it is considered that while theoretically this feature may have some value, at the present time the effect on the vision due to glare more than offsets the benefits.

Further studies will be necessary before this type of installation can be recommended.

[(388 .113 (.494) & 621 .33 (.494)]

Effect of electrification on the budget and on the balance sheet of the Swiss Federal Railways for 1927.

Although electric traction has already made considerable progress, it is often very difficult to evaluate the economic side of the question. This fact is due to the lack of accurate information on lines already in service. It will be also understood that it is as a rule difficult to make a comparison where steam and electric traction exist side by side.

We therefore believe that it may be of interest to give our readers the following note which is attached by the General Management of the Federal Railways to the balance sheet for 1927.

The working budget and the balance sheet for 1927 compared with what they would have been if none of the Federal Railway system had been electrified and that only steam power were used.

From the remarks which precede the building programme for 1927, it will be seen that by the end of the said year electric traction will be installed over 1 460 km. (907 miles) including the Seetal Railway, or in other words, on about half of the Federal Railway system; but the length of line operated by electric traction will not be more, as an average over the year, than about 1 150 km. (714 miles), representing 40 % of the system. As the electrified lines and those which have still to be electrified include those portions of line on which traffic is the heaviest, electric locomotives will haul, in 1927, 65 %, in round figures, of the total gross ton-kilometres. The very considerable development of electric traction on our system has naturally suggested the idea of estimating what operating costs would be in 1927 if steam traction were still in use over the whole of the system.

In tables I to IV which follow, we shall compare the figures for 1927 for the two methods of traction; we shall then comment on the differences which appear and, finally, we shall summarise, in the form of conclusions, the facts which have become evident.

TABLE I.

Working expenses.

	Figures for 1927 (see p. 66 of the accounts).	Figures for 1927 if steam traction were in use over the whole system.	Increase or decrease in costs if steam traction were in use over the whole system.	Remarks.
	Francs.	Francs.	Francs.	
I. General administration	7 853 140	7 853 140	...	
II. Upkeep and inspection of the permanent way	37 455 800	37 719 800	+ 264 000	1
III. Traffic Department costs	102 527 000	104 207 000	+ 1 680 000	2
IV. Motive power and workshop expenses	99 385 380	115 506 080	+ 16 120 700	3
V. General expenses	27 801 950	29 016 950	+ 1 215 000	4
Total I to V.	275 023 270	294 302 970	+ 19 279 700	
VI. Expenditure and receipts not classified at the time of closing the accounts.	—4 026 600	—4 026 600	...	5
Nett working expenses.	270 996 670	290 276 370	+ 19 279 700	

Remarks.

Upkeep and inspection of the permanent way
(see pages 56 to 59 of the budget.)

1. — From the investigation which has been made by our permanent way engineer, it is found that the upkeep of the permanent way, excluding conductor lines, costs almost 350 fr. less per kilometre for electric traction than for steam traction. It should, it is true, be taken into account that electric traction produces a greater amount of wear on the outer rail on sharp curves and that, generally speaking, very much greater stresses are imposed on the track on account of the greater weight of the locomotives; but this disadvantage, which moreover would have soon arisen with steam locomotives in view of their ever increasing weight, is much less important than the advantages, among which may be mentioned the much longer life of the lining of tunnels, the very great increase in the amount of work performed by gangs of men working in tunnels, the reduction in the maintenance work and the longer life of roofs, bridges buildings, etc. In 1927, these advantages from electric traction could not produce their full effect, and we have considered that the economies realised this year in permanent way upkeep will not exceed two-thirds of 350 fr. per kilometre. If only steam traction were employed, the expenses of upkeep and renewals (Schedule II. C. 1 and 2) would therefore be 264 000 fr. greater than that shown in the statistics.

Train working
(see pages 60 and 61 of the budget.)

2. — In comparing the statistics dealing with traffic and the output of the staff at the time when only steam traction was employed with the present day conditions, and taking into account the fact that the eight-hour day was introduced a few years ago, it will be noted that the estimated cost in the accounts for 1927 for train staff would have been at least 1 680 000 fr. more if only steam traction were employed. In 1924, although electrification was far less extended than it now is [8 400 000 train-kilometres (5 280 000 train-miles) worked electrically in 1924 against 19 000 000 (11 800 000 train-miles) in 1927] the traffic department in this case estimated the difference as 970 000 fr.

Motive Power Department and workshop expenses
(see pages 62 to 65 of the budget).

3. — If only steam traction were employed, motive power and workshop expenses would have been 16 120 700 fr. higher. This total is made up as follows :

a) *Schedules IV. A. 2 and IV A. 3* : Cost of motive power department staff and for staff employed in coaling and providing feed water and for cleaning rolling stock :

+ fr. 10 425 000

From the statistics dealing with traffic and the output of the staff, it will be seen that, if one takes into account the introduction of the eight-hour day, the estimated costs under the schedules in question for 1927 would be about 10 425 000 fr. higher, in the case of steam traction being exclusively used under the same conditions as obtained when electrification was introduced. This difference is explained by the fact that the electric locomotives are much more powerful than the steam locomotives. They haul trains about 40 % heavier and run at a higher speed. The footplate staff can therefore work on the average a considerably larger number of locomotive-kilometres and operate a greater gross tonnage than was the case with the older system of traction.

In this amount of 10 425 000 fr. are also included 2 000 000 fr. representing the cost which would be incurred by the motive power department in handling locomotive coal if all the trains were run by steam power.

b) *Schedule IV. C. 1 :* Cost of fuel :
+ fr. 17 100 000

For the number of ton-kilometres representing the traffic on which the 1927 accounts are based, the coal consumption for steam traction exclusively would be about 450 000 t. more. As a ton of coal delivered at the frontier costs about 38 fr., the increased expense would be 17 100 000 fr.

c) *Schedule IV. C. 2 :* Cost of electrical energy . . . . . — fr. 15 106 000

If there were no electric traction, it goes without saying there would be a saving of the cost of electrical energy for working the trains. In this cost is included 12 928 300 fr. which appear in the accounts for the gross output of the power stations (see remark 6). There only remains under schedule IV. C. 2 about 50 000 fr., representing the cost of energy necessary to charge the batteries of accumulator locomotives, which costs are independent of electrification.

d) *Schedule IV. C. 3 :* Cost of lubricants :
+ fr. 119 000

If only steam traction were used, the cost of lubricants would be higher for two reasons, in the first place, because according to statistics obtained in 1925, the consumption of lubricant per locomotive-kilometre is about 50 % greater for steam traction than for electric traction, and moreover because for the same traffic in gross ton-kilometres a considerably less number of locomotive-kilometres are necessary in the case of electric traction.

e) *Schedule IV. C. 5 :* Costs of cleaning and disinfecting materials, water, sand and miscellaneous . . . . . + fr. 593 000

This figure is determined from the difference between present day costs and pre-war costs, taking into account the increase in prices.

f) *Schedule IV. A. 4 and IV. D :* Switch-boards and electric lines. In the case of steam traction, there would be no expenses for switch-board staff nor for the upkeep and renewal of contact lines : — fr. 1 576 300

The upkeep and operation of sub-stations and transmission lines are included in the cost of electrical energy.

g) *Schedule IV. E :* Upkeep and renewal of rolling stock. If only steam traction were used in 1927, the maintenance costs of electric locomotives and rail cars (schedule IV. E. 2, 3 and 5) would not appear in the accounts ; on the other hand the upkeep of steam locomotives (schedule IV. E. 1) and also for carriages and wagons (schedule IV. E. 4, 6, 7, 8 and 9) would be higher, so that there would be an increased expenditure of
+ fr. 4 566 000

Increased cost under schedule IV, total	+ fr. 32 803 000
Decreased cost under schedule IV, total	— fr. 16 682 300
Nett increase. . .	+ fr. 16 120 700

Miscellaneous expenses.

4. — If only steam traction were used, the demurrage costs of rolling stock (schedule V. A. 2) should be estimated at about 225 000 fr. more by reason of the very high demurrage cost which had to be paid on foreign coal wagons.

Moreover, the contributions to the pensions and sinking funds (schedule V. C. 7 a) would have to be increased by about 990 000 fr., in view of the considerably higher cost of traffic, locomotive and station staffs.

In all, the expenses under schedule V would have been increased by 1 215 000 fr. in 1927, if steam traction were used.

5. — « Expenses and receipts not classified at the closure of the accounts » are not affected by any modification.

Table II.

Balance of accounts.

	Figures for 1927.	Figures for 1927 if steam traction were in use over the whole system.	Increase or decrease in costs if steam traction were in use over the whole system.	Remarks.
	Francs.	Francs.	Francs.	
Total receipts in respect of working . . .	390 664 600	377 736 300	— 12 928 300	6
Total working expenses	270 996 670	290 276 370	+ 19 279 700	7
Nett receipts. . .	119 667 930	87 459 930	— 32 208 000	

Remarks.

6. — For steam traction, receipts in respect of working (schedule IV. B. 3) would be, in 1927, reduced by the gross output of the power stations, that is to say, 12 928 300 fr. (see remark 3 c).

7. — This difference is shown in table I.

Table III.

Balance sheet.

	Figures for 1927.	Figures for 1927 if steam traction were in use over the whole system.	Increase or decrease in costs if steam traction were in use over the whole system.	Remarks.
Receipts.	Francs.	Francs.	Francs.	
1. Surplus of traffic receipts.	119 667 930	87 459 930	— 32 208 000	8
2. Interest from capital employed in new works	2 350 000	1 100 000	— 1 250 000	9
3. Interest on loans and deposit accounts. .	1 025 000	1 025 000	...	
4. Profit from ancillary works	4 010	...	— 4 010	10
5. Deductions from special accounts . . .	14 667 000	14 517 000	— 150 000	11
6 and 7, together	16 060	16 060	...	
Total. . .	137 730 000	104 117 990	— 33 612 010	
Expenditure.				
1. Rental on lines held on lease	47 000	47 000	...	
2. 3 and 4. Interest on consolidated stock, current debts, financial expenses, depreciation of stock, stores, together . . .	112 157 000	87 130 000	— 25 027 000	12
5. Losses in conducting ancillary works . .	120 400	120 400	...	
6. Sinking fund	16 020 000	15 450 000	— 570 000	13
7. Payments to special funds.	19 718 100	13 418 100	— 6 300 000	14
8. 9. and 10. Other expenses and depreciation, war-time deficit.	1 617 500	1 617 500	...	
Total. . .	149 680 000	117 783 000	— 31 897 000	

Remarks.

8. — Difference is shown in table II.

9. — If only steam traction were used, this amount would be 1 250 000 fr. less, this being the sum represented by the interest on money for electrical installation.

10. — If only steam traction were used, the gross receipts of the Piotta funicular railway, a small sum of 4 010 fr. would not enter into the account because this ancillary entreprise would not have been built, except for electrification.

11. — If only steam traction were used, the « deductions from special accounts » in the receipts side of the account of profits and losses would be 150 000 fr. less, in 1927, by the amount corresponding to the renewal of electrical contact lines.

12. — If only steam traction were used, the expenditure for interest, financial expenses and depreciation in stock, would be about 25 027 000 less in 1927. This figure is made up as follows :

The electrical installations, which will be in operation in 1927, and the electrical rolling stock represent a capital of about 525 000 000 fr. From this sum must be deducted 35 000 000 fr. already partly paid (value at the end of 1927) of the federal grant of 60 000 000 fr. for more rapid electrification, so that 490 000 000 fr. remain chargeable to the Federal Railways.

However, without electrification, one would have had since 1917, that is to say during the course of the last ten years, to have bought new steam locomotives costing a sum which would not have been less than 50 000 000 fr., which reduces the debt which the Federal Railways have to carry, in 1927, as a result of electrification to 440 000 000 fr.

Electrification loans which have been issued up to the present and which also serve for the conversion of loans arriving at maturity, actually cost the Federal Railways including all expenses (issue, brokerage, etc.), a little less than 5 1/2 %, so that the increased debt of 440 000 000 fr. we have already mentioned is equivalent to an annual expense of 24 200 000 fr. This sum increased by 1 250 000 fr. for the interest on capital employed in electrical equipment, as yet unfinished, which expense is dealt with in remark 9, is thus raised to 25 450 000 fr.

If only steam traction were used, it would be necessary on the other hand, in 1927, to hold a very much larger stock of coal, which would have given rise to an increased interest of 423 000 fr., from which it will be seen that, finally, the increase in interest charges due to electrification, in 1927, would only be 25 million 27 000 fr. In other words, supposing only steam traction were used, the expenditure shown in the accounts for 1927 for interest on consolidated stock and current debts, together with financial expenses, depreciation of stock, stores, etc., which amount to 112 157 000 fr would only have been 87 130 000 fr.

13. — With steam traction, depreciation costs would also be 570 000 fr. less in accordance with the Federal Railways' depreciation allowance.

14. — The payments to be made to the special funds would also have been less if all the trains were steam hauled, and this difference, corresponding to the reserves for the electrical plant, would be 6 300 000 fr.

In 1927, depreciation and payments to the reserve funds therefore represent 1.5 % of the capital expended on the plant necessary for electrical operation. In accordance with the sinking plan adopted by the Federal Railways, this sum would increase as the interest decreases relatively to the amount of the paid annuities.

Table IV.

Balance of accounts of profits and losses.

	Figures for 1927 (see pages 70 and 71 of the budget).	Figures if only steam traction were used.
	Francs.	Francs.
Total receipts	137 730 000	104 117 990
Total expenditure	149 680 000	117 783 000
Surplus of expenses	11 950 000	13 665 010
Additional costs if only steam traction were used	1 715 010	...
	13 665 010	13 665 010

CONCLUSIONS.

The accounts for operating the Federal Railways for 1927 are therefore, in round figures, 1 700 000 fr. more favourable than would be the case if only steam locomotives were employed. This result may be considered as satisfactory, especially if one considers that a large amount of electrical installation now worked was constructed during the war at extremely high prices and at a time when money was dear, relatively to which the price of coal has been reduced to a much greater extent. Moreover, the adoption of electric traction has given our Administration and to the whole country, a number of advantages which cannot be expressed in these comparative accounts.

However, to avoid drawing erroneous conclusions from these figures, it should be pointed out that up to the present electrification has only been adopted on lines carrying heavy traffic, and that the favourable results shown in our comparative figures are very largely due to this circumstance and also to the federal grant which was made in order to relieve unemployment.

Berne, 8 October 1926.

General Management
of
the Federal State Railways.

MISCELLANEOUS INFORMATION

[62. (01 (.42) & 625 .212 (.42)]

1. — Magnetic axle-testing device for « Underground » service.

Designed to facilitate detection of hair-cracks and other incipient flaws.

Fig. 1, p. 351.

(*Railway Gazette.*)

Flaws in axles are a source of considerable danger in railway operation, and one against which very stringent precautions have to be taken. However careful and frequent inspection may be, there are certain types of defects, such as hair-cracks, which cannot be readily detected by the usual methods of examination. Although hair-cracks may not be actually dangerous in the early stage, the exigency that they may develop and lead to serious trouble must always be guarded against.

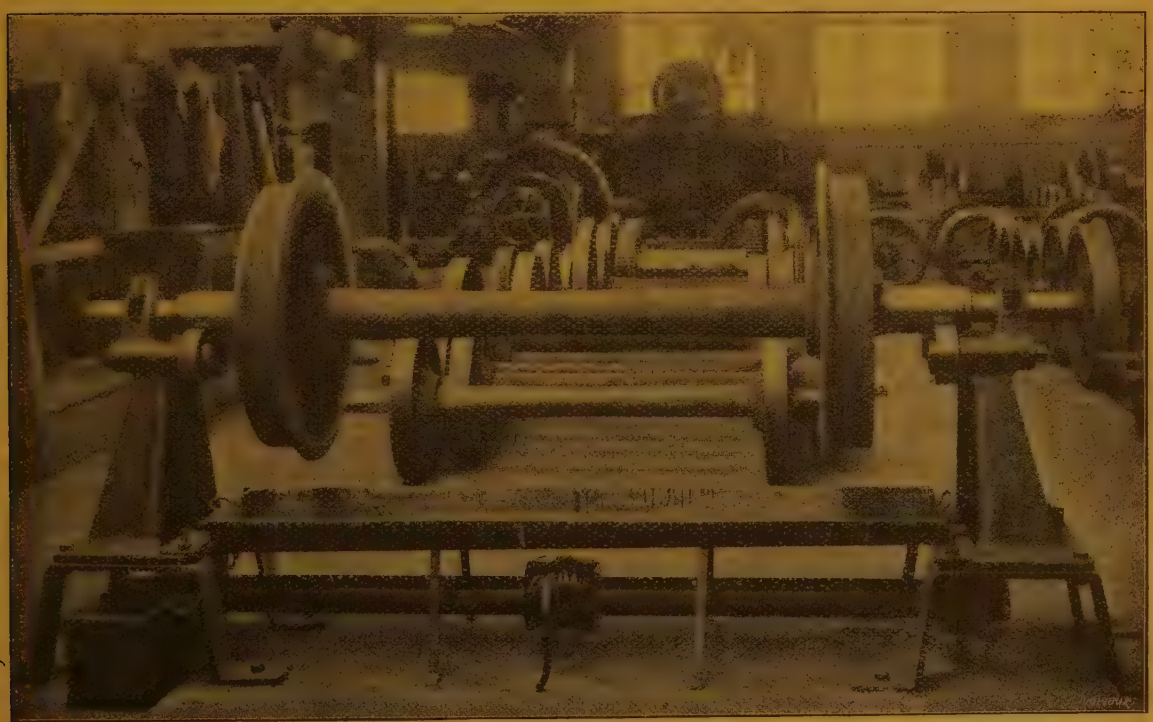

Fig. 1.

To facilitate the detection of hair-cracks in the axles of « Underground » rolling-stock a new magnetic axle-testing device has recently

been installed at the Acton Overhaul Works. It consists of two cast-iron frames and a tie-bar upon which the energising coils are placed. The axle is mounted on brass rollers, two face plates being locked flush with the journal ends. Upon energising the coils a closed magnetic circuit is formed by way of the tie-bar, the side frames, and the axle. Any flaw that is otherwise undetectable in the axle at once becomes discovered, owing to its imparting a tendency for a magnetic potential difference to be set up across the fracture wherever the latter cuts the prevailing flux. A mixture of fine iron filings and paraffin is poured over the axle, and the filings adhere to the crack and become piled up in a thick black line that is easily discernible.

It is now the practice at Acton Works to test all axles for hair-cracks as soon as they are sent in for periodical overhaul, and before operations such as tyre-turning are carried out on the wheels. This obviates any chance of an axle with hair-cracks being put back into service, only to be scrapped soon afterwards, when the cracks have developed sufficiently for them to be detected in the course of the three-weekly examination to which all axles on the « Underground » are subjected.

[621 .132.3 (.73)]

2. — The largest passenger locomotive,

By E. C. POULTNEY, O. B. E.

Figs. 2 and 3, pp. 353 and 354.

(From *The Engineer*.)

The locomotive which forms the subject of this article has been built at the Mt. Clare shops of the Baltimore & Ohio Railway from the designs of Mr. G. H. Emerson, chief of motive power and equipment, and is the largest locomotive ever constructed for passenger service, weighing with tender in working order no less than 659 000 lb., or 294.2 long tons.

The Baltimore & Ohio has a particularly difficult length of road passing over the Allegheny Mountains, known as the Cumberland Division, extending for 533 miles between Cumberland and Grafton, over which the gradients to be faced running in either direction are of a very severe character. Travelling from Cumberland, the line rises practically all the way to the summit at Altamont, 235 miles from the start, the last 80 miles being on rising gradients varying from 2.08 to 2.28 %. From Altamont to Terra Alta, about 100 miles, the line falls by 1 % gradients for 50 miles and rises for the remaining distance at rates varying from 1 to 0.48 %, after which there comes a sharp drop to Rowlesburgh, 397 miles from Cumberland, on gradients of 1.5 to 2.2 %. The next 22 miles or thereabouts rise at about 2.01 % and the last 114 miles fall at rates which vary from 2 % for 20 miles and 0.9 to 0.3 % for the remainder of the journey. It may be mentioned that the undulating tableland extending between Altamont and Terra Alta is from 2 700 to 2 550 feet above sea level. The maximum gradient is between Terra Alta and Rowlesburgh, at about the 345th mile post from Cumberland, at which point the line falls at 2.6 %, 137 feet per mile, or 1 in 37.5. For some time past heavy passenger trains have been operated over this section by *Pacific* type engines double headed, but the locomotive illustrated in diagram form in figure 2 has been designed to handle them unaided. It can haul over this section seven car trains having a total weight of 1 093 000 lb., or 488 long tons, which is the maximum load taken without assistance. Keeping in mind the weight of the locomotive, which is 294 tons, as against that of the trailing load 488 tons, an idea is obtained of the comparative size of the engine required when heavy gradients have to be negotiated. Of the total locomotive weight, 400 000 lb., or 178 tons, comprises the engine portion. This is equal to 36.5 % of the train

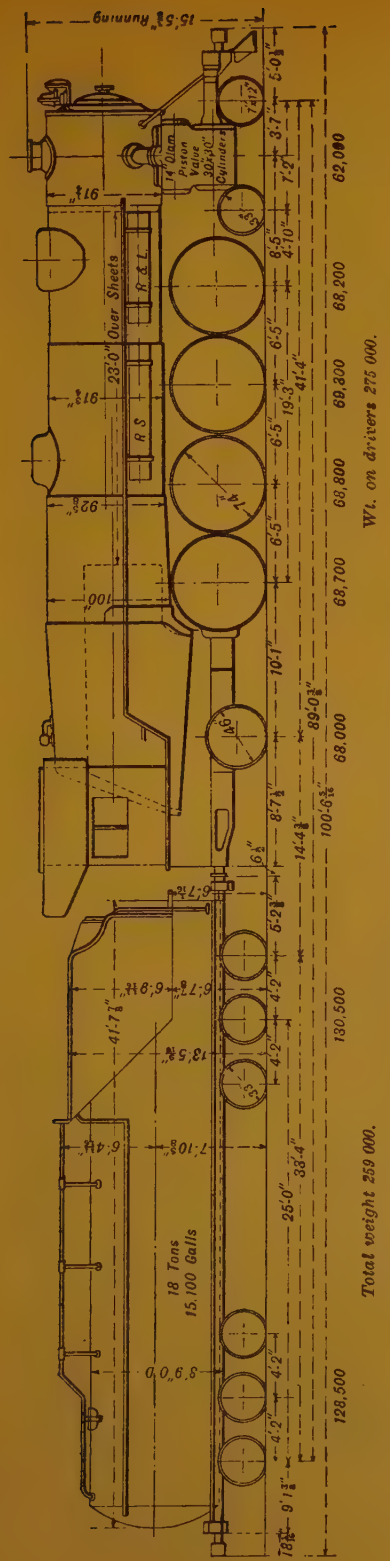

Fig. 2. — Profile and weight diagram.

weight. The tender alone, with its 36 000 lb. of coal and 15 000 gallons of water, turns the scale at 259 000 lb., or 115.5 tons, which is equal to 23.7 % of the train weight. For working over gradients of this nature the locomotive forms 37 % of the total moving load, or, put in another way, for each ton of weight in the cars there is required 0.6 ton of locomotive weight.

In order to bring out the probable performance of this locomotive on a gradient of 2.2 % the diagram (fig. 3) has been prepared. It shows the traction and train resistance curves. The upper curves indicate the traction effort available at the rim of the drivers at speeds up to 70 miles per hour and the next lower shows the available pull at the tender draw bar between the same speeds. The next gives the resistance of the seven car train on the gradient mentioned and seems to develop the fact that the highest speed obtainable on the gradient is approximately 30 miles per hour. The remaining curve plotted indicates the balance of tractive force available for acceleration up to the point where at 30 miles per hour the available effort is counteracted by the resistance offered by the train, or what Professor A. J. Wood calls the « unbalanced » tractive effort curve (1). These characteristics show, amongst other things, the power required to run the locomotive alone, and in this connection it is of interest to note that the engine could move itself over this profile at about 60 miles per hour.

In order to bring the size of this locomotive into greater prominence the comparative data collected together in the tabulated statement giving the dimensions and proportions of previous Mountain type locomotives have been drawn up. In all, nine different examples are given, arranged in order of weight. The new Baltimore & Ohio engine easily heads the list, being 24 000 lb. heavier than the Delaware, Lackawanna Western design, which follows it. Of this excess weight 16 500 lb. are on the drivers to take care of the increased tractive effort of 13 362 lb. The column showing the cylinder horse-power gives this figure computed

(1) *Locomotive Operation*, Arthur J. Wood.

Comparative particulars of typical

RAILWAY.	Total weight.	Adhesive weight.	Diameter drivers.	Cylinders.	Steam pressure.	Tractive force.
	lb.	lb.	inches.	inches	lb.	lb.
Baltimore & Ohio	400 000	274 000	74	30 × 30	220	68 200
Delaware, Lackawanna & Western . . .	376 000	257 500	69	28 × 30	200	54 838
Great Northern	357 000	283 000	73	29 × 28	200	54 830
Atchison, Topeka & Santa Fé.	353 900	227 700	69	28 × 28	200	54 100
United States Railroad Administration . .	352 000	243 000	69	28 × 30	200	58 000
Union Pacific.	345 000	230 000	73	29 × 28	200	54 838
Norfolk & Western	341 000	236 000	70	29 × 28	200	57 200
New York Ontario & Western	317 000	208 200	69	27 × 28	200	50 300
Southern Railway	314 000	214 800	69	27 × 28	190	47 800

NOTE. — Tractive force calculated by taking

by the Cole method and is based on a piston speed of 1 000 feet per minute.

In general, the designs call for little comment, but mention may be made of one or two of them. The United States Railroad Administration is interesting, as it is the standard design of heavy 4-8-2 engine worked out during the period of Government control, and is jointly the product of the building firms and railway officers, whilst the Atchison, Topeka & Santa Fé engines, when built in 1918, were the largest of their type. This locomotive was illustrated in *The Engineer*, 13 June 1919.

The new Baltimore & Ohio engines, beyond their great size and power, follow along the lines of conventional locomotive practice in the United States, and the general features can be noted from the illustration. The full gear cut-off is 88 %. A list of the leading dimensions of this notable locomotive is appended. It gives also the estimated powers and the boiler and weight ratios. It should be noticed that the axle weight on the drivers is in the neighbourhood of 30 tons. The writer is indebted to Mr. G. H. Emerson for the photograph and weight diagram and other data, on which this article on the world's largest passenger locomotive is principally based.

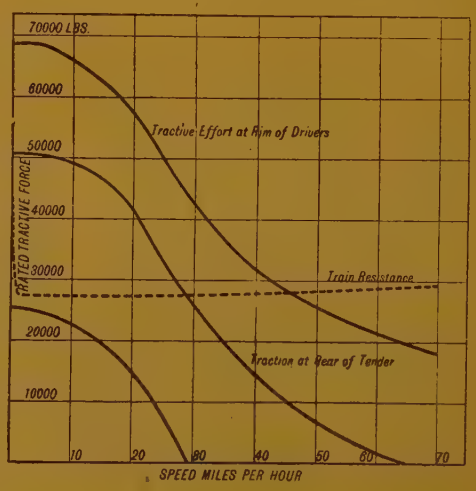

Fig. 3. — Locomotive performance curves.

General particulars.

Total weight, engine . . . . . 400 000 lb.
Adhesive weight . . . . . 275 000 lb.
Tender weight . . . . . 259 000 lb.

Mountain type passenger locomotives.

Heating surfaces.			Grate area.	Cylinder H. P.	Comb. H.S. / Grate area.	Comb. H. S. / Cyl. H. P.	Total weight / Comb. H. S.
Evaps.	Superh'r.	Combined.					
square feet.	square feet.	square feet.	square feet.				
5 591	1 305	6 896	89.17	3 568	77.2	1.93	58.2
4 974	1 242	6 216	84.0	2 824	78.0	2.20	59 8
4 971	1 368	6 339	88.0	3 030	72.0	2.12	56.3
4 792	1 087	5 879	71.7	2 824	81.9	2.07	65.5
4 666	1 085	5 751	76.3	2 824	75.3	2.04	61.2
4 974	1 242	6 216	84.0	3 030	74.2	2.05	55.4
3 984	882	4 866	80.3	3 030	60.6	1.61	71.5
3 654	1 010	4 664	66.8	2 624	69.5	1.78	68.2
3 668	942	4 610	66.7	2 624	69.2	1.75	68.2

the M. E. P. as 85 °/₀ of the boiler working pressure.

Total engine and tender. . . . . 659 000 lb.
Cylinders (two) . . . . . 30 by 30 inches.
Valves, type. . . . . Piston, 14 inch diameter.

Wheels :

Driving. . . . . 74 in. diameter.
Leading truck . . . . . 33 inches.
Trailing truck . . . . . 46 —

Boiler steam pressure. . . . . 220 lb. per sq. in.
Boiler diameter, largest outside. 100 inches.
Tubes, number and diameter . . 269—2 1/4 inches.
Flues, number and diameter . . 48—5 1/2 —
Length between tube plates. . . 22 ft. 10 3/4 —
Fire-box, size inside . . . . . 96 – by 132 1/4 in.

Heating surfaces :

Tubes. . . . . 3 626 square feet.
Flues . . . . . 1 582 —
Fire-box, including combustion chamber and tubes . . 383 —

Total evaporative. . . . . 5 591 square feet.
Superheater. . . . . 1 305 —

Total combined. . . . . 6 896 square feet.
Grate area. . . . . 89.17 square feet.

Special equipment :

Valve gear . . . . . Baker.
Reserve gear . . . . . Ragonnet « B ».
Stoker . . . . . Duplex.
Superheater . . . . . Type A.

General data estimated.

Rated tractive force, M. E. P., 85 °/₀ boiler pressure . . . . . 68 200 lb.
Cylinder H.P. (Cole) at 1 000 foot piston speed 3 568
Speed at above power. . . . . 44 miles per hour.

Ratios.

Boiler proportions :

Rated tractive force ÷ combined heating surface = 9.9.
Rated tractive force × diameter of drivers ÷ combined heating surface = 73.1.
Combined heating surface ÷ grate area = 77.3.
Fire-box heating surface × 100 ÷ combined heating surface = 5.56.
Superheater surface × 100 ÷ combined heating surface = 18.95.

Weight proportions :

Adhesive weight ÷ rated tractive force = 4.03.
Adhesive weight × 100 ÷ total weight = 69.
Total weight ÷ combined heating surface = 58.2.

3. — Statistics of the Government Railways of Japan for the year ending 31 March 1926.

(COMPARED WITH THE FOUR PRECEDING YEARS.)

ITEM.	Year.				
	1925-1926	1924-1925	1923-1924	1922-1923	1921-1922
Area (square miles)	(2) 129 247	(2) 129 247	(2) 129 247	(2) 129 247	(2) 129 247
Population	(2) 60 257 931	(2) 60 257 931	(2) 58 481 500	(2) 57 655 800	(2) 56 787 300
Miles of lines worked (road) . . . (m. ch.)	7 837.08	7 558.41	7 350.54	7 013.33	6 722.21
Single track sections . . . . . (—)	6 658.07	6 411.29	6 269.54	6 012.20	5 791.21
Double track sections . . . . . (—)	1 126.68	1 099.44	1 048.09	972.37	905.51
Triple and more track sections (—)	52.13	47.56	32.71	28.56	25.23
Total miles of main track . . . . (—)	9 115.55	8 797.19	8 500.49	8 079.03	7 711.06
Total miles of all tracks . . . . . (—)	12 619.60	12 217.40	11 735.50	11 139.24	10 670.74
Average area per mile of line . . (sq. m)	16.5	17.1	17.6	18.4	19.2
Average miles of lines per 100 000 of population.	13.0	12.6	12.7	12.1	11.8
Average miles of line worked { passenger traffic.	7 627.7	7 368	7 057.8	6 778.2	6 540.2
Average miles of line worked { goods traffic. . .	7 691.8	7 433.1	7 123.6	6 845.5	6 610.2
Number of stations	2 292	2 207	2 134	2 027	1 940
Capital (*yen*)	2 500 154 609	2 323 210 254	2 149 783 769	1 978 915 868	1 779 569 007
Capital per mile of line (*yen*)	319 016	307 384	292 460	282 162	264 731
Working revenue (*yen*)	480 450 904	470 927 240	443 354 781	429 593 995	397 599 494
Working expenses (*yen*)	259 439 720	266 232 608	261 242 250	229 778 431	218 650 405
Profit (*yen*)	221 011 184	204 694 632	182 112 531	199 815 564	178 949 089
Add : Appropriations from the reserve account.	...	...	...	...	...
Deduct : Control of private railway and survey expenses	334 310	376 775	353 771	343 226	342 678
Deduct : Additional works expenses	5 685 453	6 218 605	6 386 148	8 543 124	8 253 212
Deduct : Interest charges	67 931 932	65 295 103	62 857 737	57 981 722	53 719 899
Deduct : Subsidies to private local railways . .	4 152 609	2 988 770	2 218 875	1 564 427	914 066
Add or *deduct* (△) : Balance on the suspense account	351 807	3 956	△ 31 905	146 303	△ 181 211
Balance (net profit)	143 258 687	129 819 335	110 264 095	131 529 368	115 538 023
Percentage of expenses to revenue	54.0	56.5	58.9	53.5	54.9
Percentage of profit to capital	9.6	9.5	9.2	11.2	11.6
Working revenue per mile (*yen*)	62 415	63 297	62 055	61 253	59 355
Working expenses per mile (*yen*)	33 703	35 784	36 566	32 762	32 527
Profit per mile (*yen*)	28 711	27 513	25 490	28 491	27 050
Working revenue per train-mile (*yen*)	5.18	5.18	5.28	5.25	5.19
Working expenses per train-mile (*yen*)	2.80	2.93	3.11	2.81	2.86
Profit per train-mile (*yen*)	2.38	2.25	2.17	2.44	2.34

Locomotives					
Passenger carriages	10 308	10 053	9 493	9 298	8 575
Goods wagons	59 607	57 882	56 810	55 405	52 492
Aggregate weight of locomotives (*ton*)	274 716	271 800	259 110	242 600	227 809
Aggregate number of seats of carriages	551 451	523 046	477 035	460 761	408 514
Aggregate loading capacity of wagons	744 032	712 559	692 362	664 711	614 330
Employees	195 876	195 555	188 783	180 860	163 371
Monthly compensation of employees (*yen*)	10 062 833	9 883 003	9 628 261	9 286 921	8 796 234
Number of passengers carried	677 085 503	635 454 260	576 472 225	509 808 705	454 535 924
Number of passengers carried one mile	11 645 130 433	11 250 551 062	10 669 134 204	9 760 776 688	8 897 704 223
Passenger-miles per mile	1 526 690	1 488 404	1 511 608	1 391 732	1 367 994
Passengers per train-mile	206.2	209.2	216.1	206.0	207.3
Passengers per vehicle-mile	14.1	14.3	15.0	14.5	14.6
Average miles of journey per passenger	17.2	17.7	18.5	19.0	19.6
Passenger earnings (*yen*)	225 855 164	222 235 788	214 570 149	199 750 653	145 765 677
Passenger earnings per mile (*yen*)	29 610	29 401	30 402	28 481	28 561
Passenger earnings per passenger per mile (*yen*)	0.0194	0.0198	0.0201	0.0205	0.0209
Coaching receipts (*yen*)	261 787 479	258 810 661	249 470 408	232 174 781	214 519 192
Coaching receipts per mile (*yen*)	33 404	34 256	35 347	33 155	32 982
Coaching receipts per train-mile (*yen*)	5.16	4.14	5.70	5.59	5.63
Tonnage of goods hauled	71 939 246	70 057 345	64 782 436	64 070 573	57 394 029
Tonnage of goods hauled one mile	7 226 686 969	7 047 680 468	6 392 328 633	6 364 956 806	5 888 701 225
Ton-miles per mile	939 531	932 381	897 345	907 542	890 866
Tons per train-mile	199.3	190.4	184.7	184.3	175.1
Tons per vehicle-mile	6.3	6.1	6.1	6.1	4.4
Average miles of goods hauled per ton	100.5	100.6	98.7	99.3	102.6
Goods earnings (*yen*)	197 169 594	193 030 389	176 564 429	177 800 802	166 034 174
Goods earnings per mile (*yen*)	25 633	25 537	24 786	25 352	25 059
Goods earnings per ton per mile (*yen*)	0.0273	0.0274	0.0276	0.0279	0.0282
Goods wagon receipts (*yen*)	198 786 210	194 563 653	178 109 089	179 220 166	167 241 648
Goods wagon receipts per mile (*yen*)	25 843	25 740	25 003	25 554	25 301
Goods wagon receipts per train-mile (*yen*)	6.06	5.22	5.76	5.95	5.85
Passenger train mileage	(1) 50 696 543	(1) 48 409 041	(1) 43 755 981	(1) 41 538 520	(1) 38 099 966
Goods train mileage	32 294 876	33 339 255	30 874 570	30 144 228	28 573 173
Mixed train mileage	9 261 218	9 063 209	9 340 972	10 100 230	9 871 377
Total train mileage	92 752 637	90 811 505	83 971 523	81 782 978	76 544 516
Locomotive mileage	107 794 646	106 492 678	100 375 722	96 683 544	91 559 302
Passenger car mileage	826 005 408	787 432 208	710 303 375	674 922 715	609 996 099
Goods wagon mileage	1 155 872 082	1 147 050 092	1 042 313 720	1 036 281 569	1 370 576 836
No. of vehicles coupled per train — passenger car	14.6	14.6	14.4	14.3	14.2
No. of vehicles coupled per train — goods wagon	31.9	31.0	30.1	30.0	40.8
No. of vehicles drawn per locomotive	24.74	24.74	23.43	23.29	22.57

N. B. — (1) Includes the mileage travelled by electric trams and steam motor cars converted at the rate of 10 cars = one train.
(2) Does not cover Taiwan, Chosen and Japanese Karafuto.

[625 .245 (.73)]

4. — 70-ton hopper cars for the Delaware, Lackawanna & Western Railroad.

Figs. 4 and 5, p. 359.

(*Railway Age.*)

In order to facilitate the handling of coal traffic, the Delaware Lackawanna & Western recently retired about 400, 40-ton steel and wood coal cars and replaced them with modern 70-ton hopper-gondola cars which were built by the American Car & Foundry Company at its Berwick, Pa., plant. These new cars are the first of this capacity and type to be used by the Lackawanna. A great part of the coal handled by the Lackawanna is anthracite and in building these cars particular attention was directed to the construction of the car body and doors in such a manner as to eliminate, as far as possible, the loss of the small sizes of coal by sifting.

Increased loading capacity was obtained in this car by arranging the stakes on the inside of the side sheets instead of on the outside, as is the usual practice. There are 10 pressed steel stakes on each side of the car, two of which are secured to the center construction by means of deep gusset plates in order to prevent the bulging of the sides. The sides are further reinforced by four pressed steel cross ties placed at the top of the sides and properly spaced to give uniform stiffness. Three double hoppers are used which provide six extra large door openings with independently operated hopper doors to facilitate easy and rapid discharge of the lading. The details of the hoppers, such as the side and center hopper sheets, stiffeners, door spreaders and hopper doors, are standardized and interchangeable in order to minimize the number of parts required for repair work. The principal dimensions of the cars are shown in the accompanying table.

Principal dimensions of the Delaware Lackawanna & Western Railroad 70-ton cars:

Length over pulling face of couplers	43 ft. 11 1/2 in.
Length over buffers	41 ft. 5 in.
Length inside, in the clear	40 ft. 0 in.
Width, over all	10 ft. 2 3/4 in.
Width inside	10 ft. 1 in.
Height from top of rail to top of body	10 ft. 8 in.
Height from top of rail to hopper door opening	10 5/8 inches.
Center to center of trucks	31 ft. 5 in.
Wheel base of trucks	5 ft. 8 in.
Capacity, level full	2 755 cubic feet.
Capacity, with 10-inch average heap	3 090 —
Light weight	50 800 lb.

Details of construction.

The under-frame is of the built-up type with two center sills composed of A. R. A. standard 12-inch special rolled channels, weighing 40.3 lb. per foot, extending the entire length of the car, with a 1/2-inch by 12 3/4-inch top cover plate. The ends of the channel are coped away to receive the buffer attachments together with a Union coupler centering device. There are seven pan-shaped 1/4 inch pressed fillers, and one saddle of 1/4 inch pressed steel located under each longitudinal ridge sheet. The body bolsters are built up of plate and angle construction. The bolster and side connection angles extend into the car. Diagonal braces of 5-inch by 3 1/2-inch by 3/8 inch angle section are connected to the center sills and bolsters by 5/16-inch gusset plates.

The top side angle is a 5/8-inch by 3 1/2-inch by 3/8-inch bulb angle running the entire length of the car. The bottom flange angle at the sides is 5 inches by 3 1/2 inches by 3/8 inch and extends from bolster to bolster. The side sheets are 1/4-inch plate, except that at the end panels they are 3/16-inch plate, pressed with an offset to accommodate the side ladders. The side sheets are all pressed inward at the top with ten 1/4-inch stiffeners on each side of the car. The sub side-sill is

Fig. 4. — 70-ton three-hopper steel car built for the Delaware, Lackawanna & Western.

Fig. 5. — End view of the car, showing the arrangement of the hand brake equipment.

a 9-inch, 17.5-lb. channel. The end sills are 9-inch, 17.5-lb. channels with 3 1/2-inch by 3 1/2-inch by 5/16-inch angle connections to the center sills and top cover plate. The corner posts are 3 1/2-inch by 3 1/2-inch by 5/16-inch angle and the end posts are of 6-inch, 8.2 lb. channel, except at the corner where the hand brake is located. The end post at this point consists of two 4-inch, 8.2 lb. Z-bars arranged for the application of the Ajax handbrake. There are six hopper doors pressed from 1/4-inch plate, known as Ajax corrugated doors manufactured by the Union Metal Products Company. The six doors operate independently and are equipped with the Enterprise Railway Equipment Company's Type « D » door operating mechanism.

Draft gear and trucks.

The cars are equipped with the Cardwell draft gear, type G. 111-AA, secured between Universal cast steel draft lugs. The couplers, furnished by the Gould Coupler Company, are A. R. A. type « D » with 6-inch by 8-inch shank and the regular A. R. A. butt suitable for either the single key, vertical yoke or double key horizontal yoke attachment. The draft keys used are of 6-inch by 1-inch by 1/2-inch carbon steel, quenched and tempered, with Universal draft key retainers. The side bearings are of the spring controlled anti-friction type manufactured by E. S. Woods & Company.

The car superstructure is mounted on Bettendorf four-wheel trucks. These trucks have cast steel side frames with journal boxes cast integral and are arranged for six truck springs at each side frame. Cast steel truck bolsters are used, with integral center plates and they are arranged for the Barber lateral motion device. The trucks are designed for 70 tons capacity and are equipped with double coil springs manufactured by the Crucible Steel Company. Davis 33-inch steel wheels, manufactured by American Steel Foundries, are mounted on 6-inch by 11-inch axles.

The cars are equipped with the National Car Equipment Company's Ajax hand brake which is designed to develop not less than 3 950 lb. pull at the brake cylinder push rod connection to the cylinder lever. The air brakes are Westinghouse automatic quick action, schedule K. D. 1012 with K-2 triple valve, centrifugal dirt collector, and double spring retaining valve, 10-20 type. A. R. A. No. 2 plus Creco trussed brake beams with which Creco four-point brake beam supports are employed.

[621 .132.3 (.73)]

5. — Denver & Rio Grande Western Railroad 4-8-2 three-cylinder locomotives.

Figs. 6 to 12, pp. 361 to 366.

(From the *Railway Mechanical Engineer.*)

The Baldwin Locomotive Works has recently completed an order for ten three-cylinder locomotives for the Denver & Rio Grande Western. These locomotives, one of which was exhibited during the recent convention of the Mechanical Division of the American Railway Association at Atlantic City, are designed to meet the demands of unusually severe operating conditions over a road where the maximum grades are 3 % and the sharpest curves are 16°. They develop a maximum tractive force of 75 000 lb. which is exceptional for an eight-coupled design, giving these locomotives a hauling capacity that is exceeded by comparatively few locomotives in freight service.

It was the aim of the designers to obtain maximum capacity in a coal-burning unit of the 4-8-2 type, without exceeding certain specified weights and clearance limits. To accomplish this it was essential that all superfluous weight be omitted and the detail parts are consequently as light as is consistent with the required strength. The total weight of one of these locomotives is 419 310 lb., of which

Fig. 6. — Three-cylinder Mountain type locomotive built for the Denver & Rio Grande Western by the Baldwin Locomotive Works.

290 530 lb. is distributed over the drivers, 67 880 lb. on the engine truck and 60 900 lb. on the trailing truck. All three cylinders are 25 by 30 inches and the outside diameter of the driving wheels is 67 inches. The boiler pressure is 210 lb. per square inch.

A novel feature in the design is the construction of the cylinders. The three cylinders are of cast iron and cast separate from each other, the center cylinder being cast in one piece with the saddle. Both sides of the center cylinder casting are flanged to receive the two outside cylinders which are cast from the same pattern. The front frame rails are single, measuring 6 inches in width by 3 1/2 inches in depth. The outside cylinders are bolted securely to the rails and also to the center cylinder casting.

The steam chest for the inside cylinder is placed on the right hand side and receives its steam supply from a branch leading from the right hand steam pipe. The location of this steam chest makes it necessary to run the exhaust passage for the right hand and inside cylinders underneath the latter to the left side of the casting, and then up to the base of the exhaust pipe. All pipe joints subject to live steam pressure are placed outside where they are easily accessible.

The steam distribution to each cylinder is controlled by a 12-inch piston valve having a steam lap of 1 1/8 inches and an exhaust clearance of 1/4 inch. The valves are set with a travel of 6 1/2 inches and a lead of 1/4 inch, and are controlled by a Walschaerts gear. The motion on the left side is arranged in the usual way; but on the right side the main pin has attached to it a double return crank which operates two links, one for the inside cylinder and the other for the right hand outside cylinder. The inside valve receives its lead from the corresponding crosshead and the movement is transmitted from the plane of the latter to that of the outside link through a horizontal shaft which is fulcrumed in a rocker, the bearing of which is bolted to the main frames. This horizontal shaft passes through a crosshead to which the inside valve stem is attached. The three valve motions

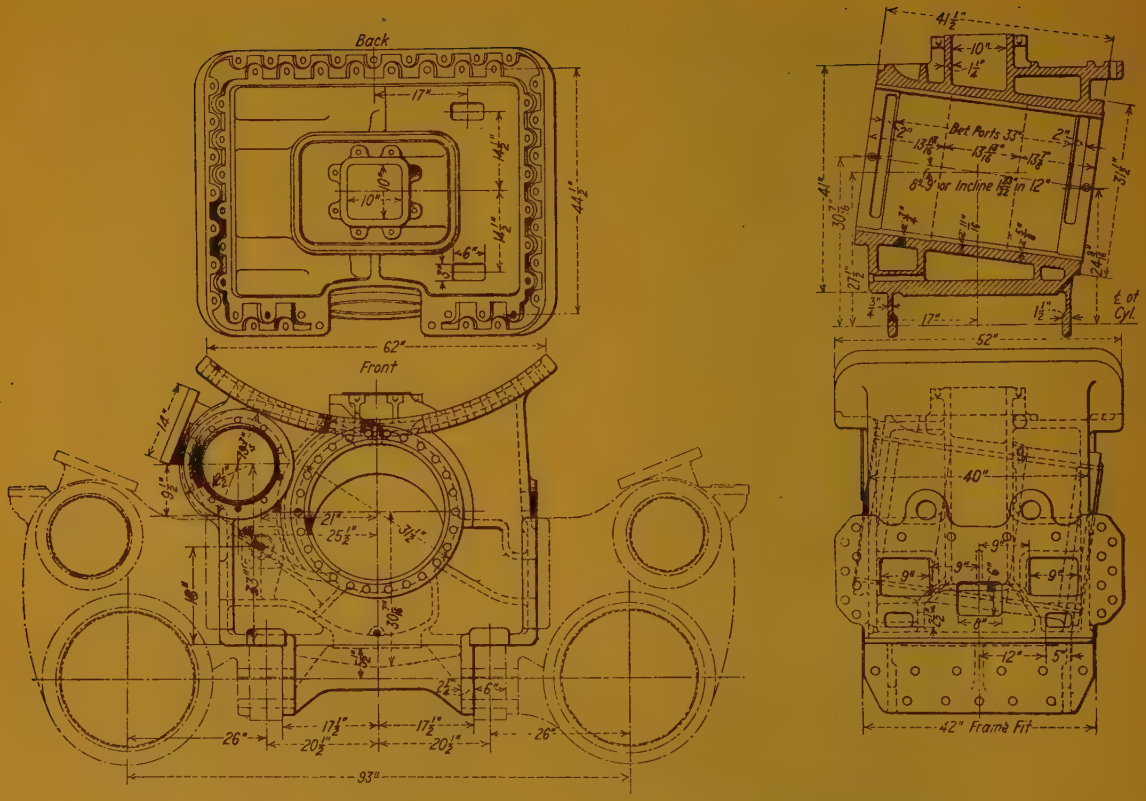

Fig. 7. — Drawing showing the construction of the inside cylinders.

are controlled simultaneously by a power reverse mechanism.

The outside eccentric rod on the right hand side has a solid end of the usual design at the back end, while the inside rod necessarily has a split brass, which is held in place by a cap secured by two horizontal bolts.

The piston of the inside cylinder is connected to the second pair of drivers and those of the two outside cylinders to the third pair. The outside crossheads are of the alligator type, while the inside one is of the underhung type, this design being used to provide sufficient clearance above the first driving axle. The outside guides and crossheads are unusually long, being 9 ft. 1/2 in. in length. The guides are supported by two bearers, placed respectively ahead of and behind the leading drivers. The lips of the inside guide, which take the pressure when backing up, are separate pieces secured to the upper shoe of the guide by vertical bolts. This form of construction is amply strong and facilitates machining the various parts.

The main and side rods are of I-section and made of carbon vanadium hammered steel, annealed. The crank pins are of the same material. Floating bushings are used on the main side rod pins and also on the back ends of the outside main rods. The back end of the inside main rod is of the strap type.

The crank axle, a product of the Standard Steel Works Company, Philadelphia, Pa., is of the built up type in five parts : two end

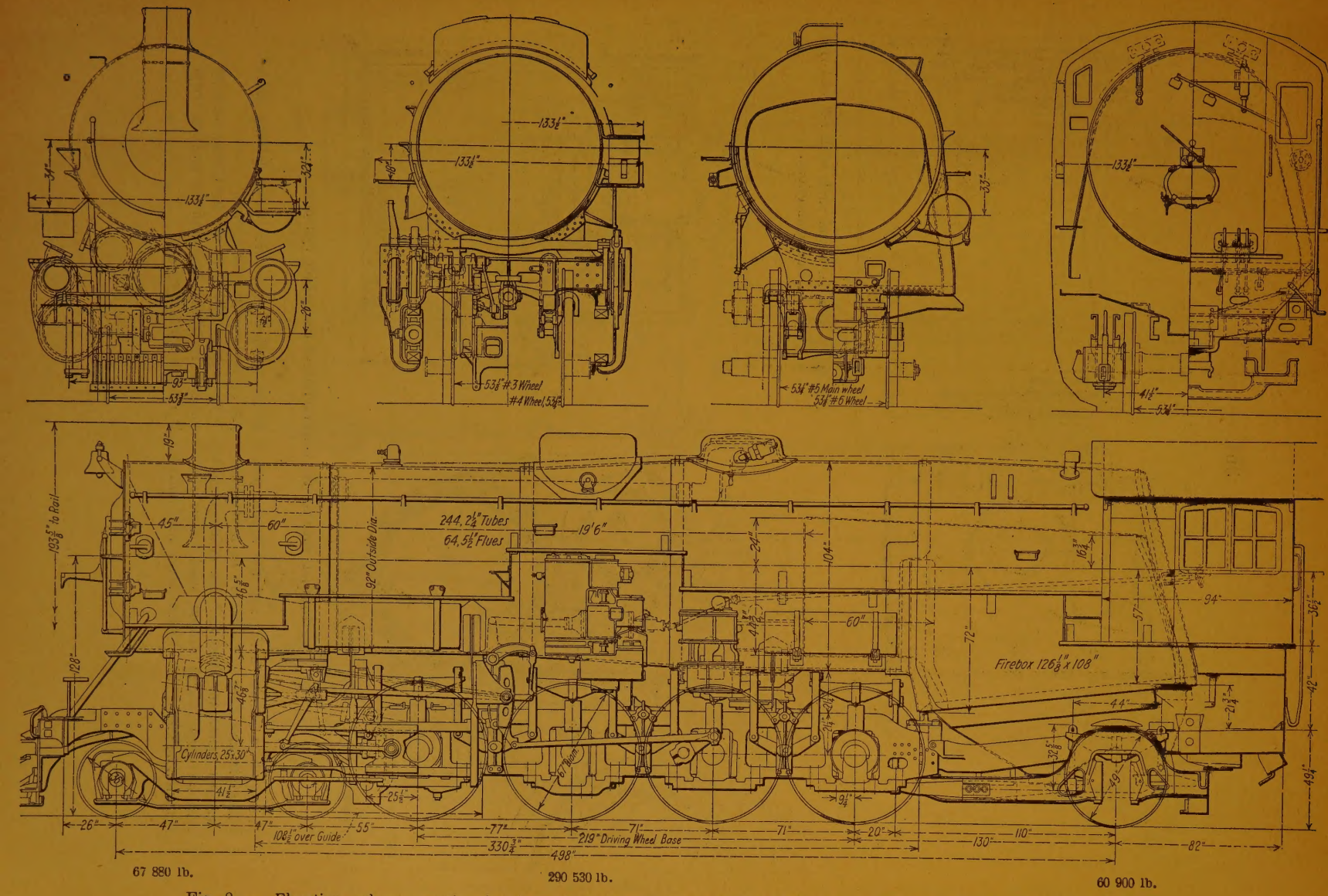

Fig. 8. — Elevation and cross section drawings of the Denver & Rio Grande Western three-cylinder Mountain type locomotive.

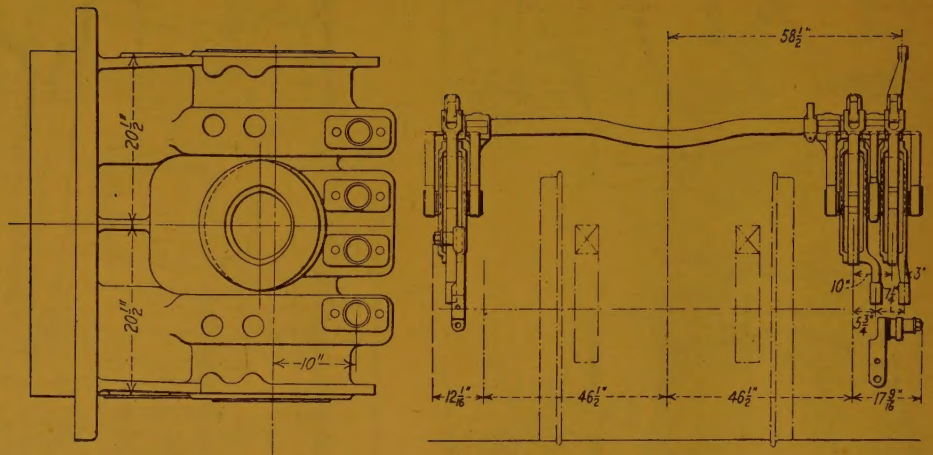

Fig. 10. — The horizontal shaft with left and right side connections.

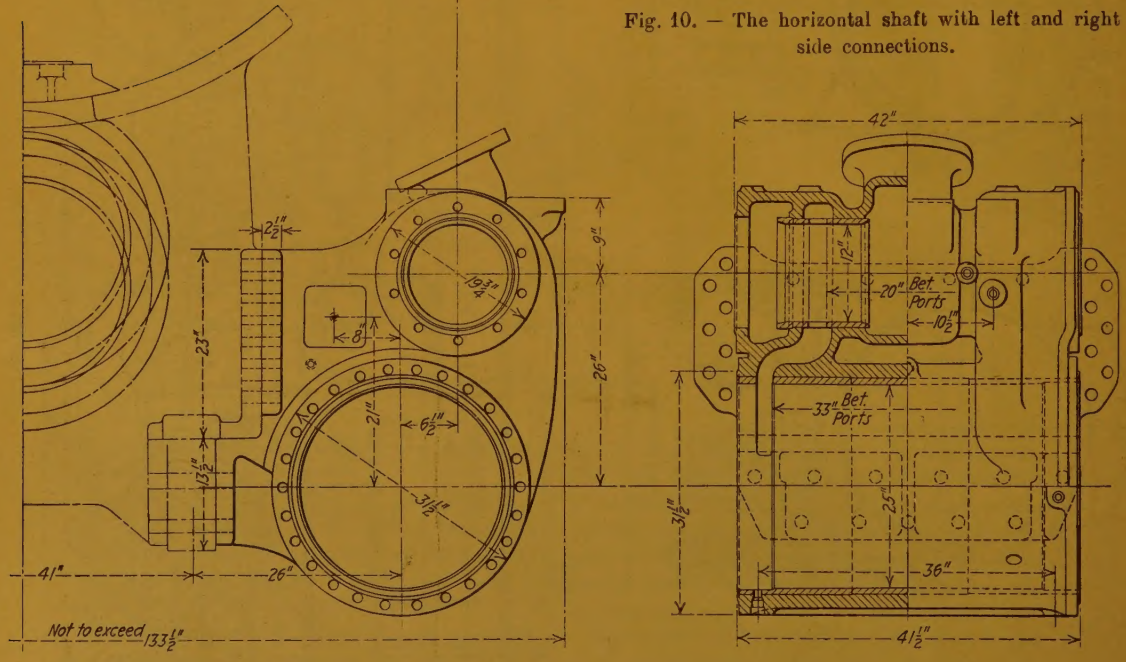

Fig. 9. — Drawing showing the construction of the outside cylinders.

pieces, two crank cheeks and the center pin. The pin and axle portions are of normalized carbon vanadium steel and the crank cheeks are of open hearth carbon steel. Cast steel counterweights, filled with lead, are bolted to the cheeks. The three other driving axles are of carbon vanadium steel.

The engine truck is of the Commonwealth constant resistance type and the trailing truck is of the Delta type.

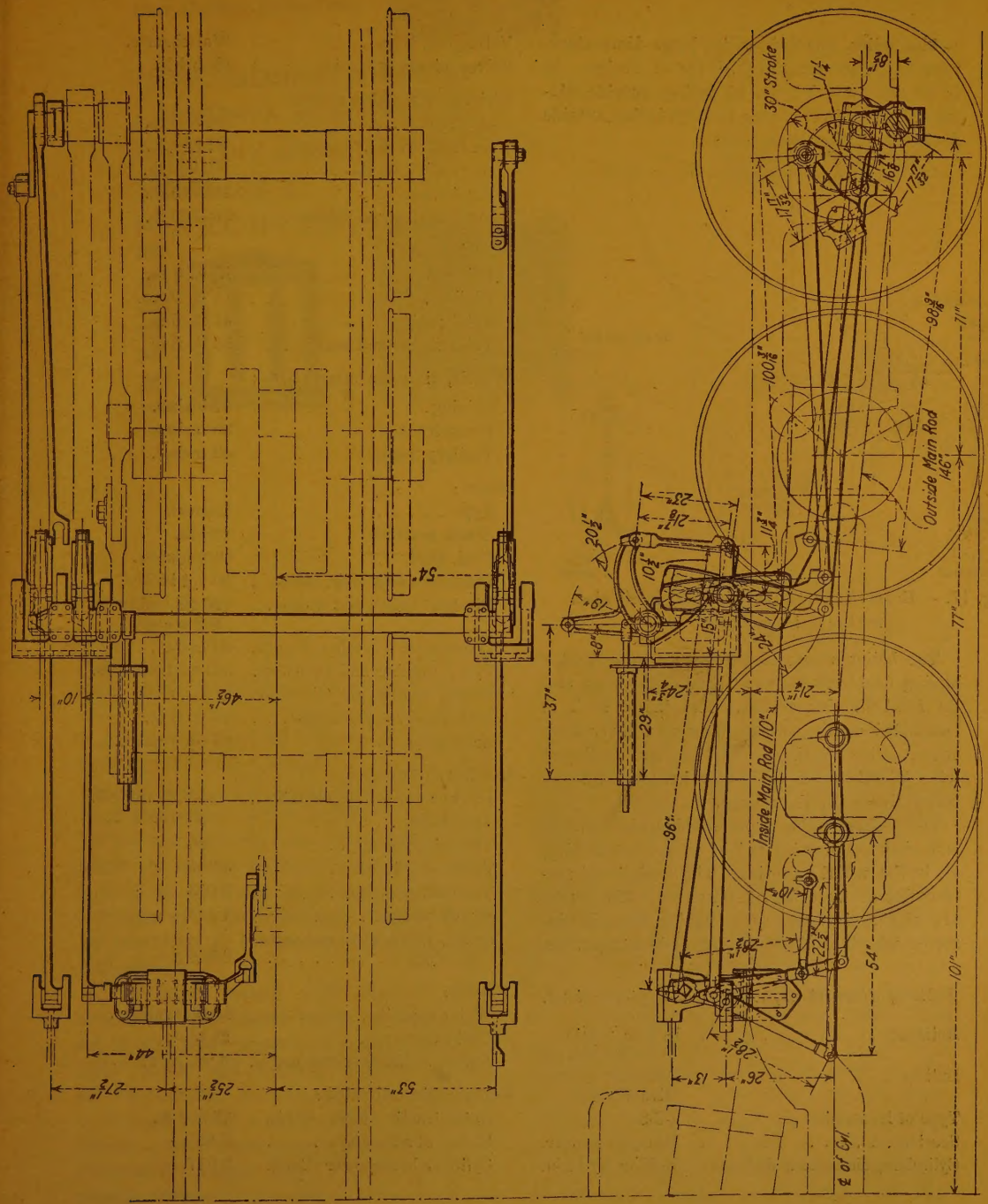

Fig. 11. — Side and plan views of the Walschaerts valve gear as designed for three-cylinder operation.

The boiler, apart from its large dimensions, represents no unusual features of design. It is of the conical type, 92 inches outside diameter at the first course and 104 inches outside diameter at the dome course.

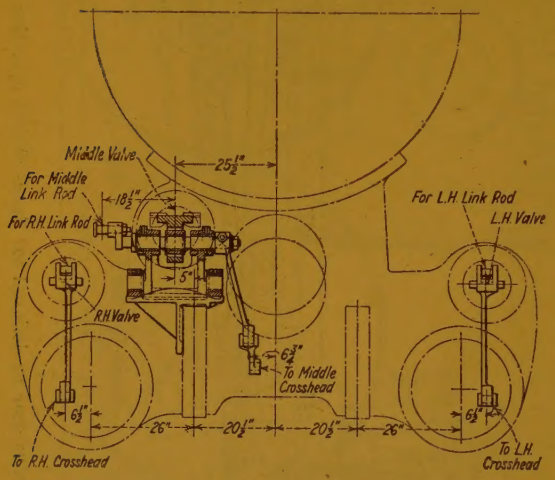

Fig. 12. — Cross section of the valve gear mechanism for the inside cylinder.

The boiler accessories include a superheater having 64 elements, two thermic syphons and du Pont-Simplex stoker. The brick arch is supported on the two syphons and three arch tubes. Five of the locomotives are equipped with Worthington feedwater heaters and the other five with the Elesco.

The stoker engine is mounted on the forward end of the tender instead of on the locomotive, as is the usual practice. The tank is of rectangular or water leg construction. The trucks are of the six-wheel type, having 36-inch diameter wheels and 6-inch by 11-inch journals.

Table of dimensions, weights and proportions.

Railroad	Denver & Rio Grande Western.
Builder	Baldwin Locomotive Works.
Type of locomotive	4-8-2
Service	Heavy passenger.
Cylinders, diameter and stroke	(3) 25 in. by 11 in.
Valve gear, type	Walschaerts.
Valves, piston type, size	12 inches.
Weights in working order :	
On drivers	290 530 lb.
On front truck	67 880 lb.
On trailing truck	60 900 lb.
Total engine	419 310 lb.
Total engine and tender	716 400 lb.
Wheel bases :	
Driving	18 ft. 3 in.
Rigid	11 ft. 10 in.
Total engine	41 ft. 6 in.
Total engine and tender	86 ft. 5 in.
Wheels, diameter outside tires :	
Driving	67 inches.
Front truck	36 inches.
Trailing truck	49 inches.
Boiler :	
Type	Conical.
Steam pressure	210 lb.
Fuel, kind	Bituminous.
Diameter, first ring, inside	90 1/4 inches.
Firebox, length and width	126 1/8 inches by 108 inches.
Combustion chamber, length.	60 inches.
Flues, number and diameter	64 — 5 1/2 inches.
Tubes, number and diameter.	244 — 2 1/4 —
Length over tube sheets	19 ft. 6 in.
Grate area	95 square feet.
Heating surfaces :	
Firebox and comb. chamber	402 square feet.
Arch tubes	25 — —
Thermic syphons	85 — —
Tubes and flues	4 581 — —
Total evaporative	5 093 — —
Superheating	1 495 — —
Comb. evaporative and superheating	6 588 — —
Tender :	
Water capacity	15 000 gallons.
Fuel capacity	25 tons.
Journals, diameter and length.	6 in. by 11 in.
General data estimated :	
Rated tractive force, 85 %	75 000 lb.
Factor of adhesion	3.88
Cylinder horsepower (Cole)	3 547